Vol. XVI. — No. 2. February 1934

BULLETIN
OF THE
INTERNATIONAL RAILWAY CONGRESS
ASSOCIATION
(ENGLISH EDITION)

625. 115]

Railway curves.

The accuracy of the method of adjusting curves by correction of the versines.

Rule for determining whether a curve was properly laid in originally.

Calculation of the ordinates of a curve in terms of its versines,

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The theory of the method of adjusting curves by correction of the versines is based on a principle which mathematically is not up to the standard of accuracy to which we are accustomed in permanent way calculations.

Consequently, when Lefort, its inventor, put it forward, certain engineers doubted its accuracy, and it is only since its application on the ground has shown that it gave better results than those obtained by other methods that railway engineers have come to regard it more favourably.

Even today, however, certain engineers, and these not the least eminent, consider it somewhat suspect through the lack of mathematical exactitude in the principle upon which it is based.

It is certainly suprising that, in order to convince them, no one has endeavoured to determine the theoretical value of the error inherent in the method, an error which for the moment is masked by errors in the measurements made upon the ground.

We propose, in this article, to calculate this error.

If two curves are to be connected, these two curves must have, at their beginning as at their ends, two common pegs. Consequently, the last two ordinates of each of the two curves ought to be equal. We are led, therefore, to calculating the ordinates of a curve in terms of the versines of this curve.

Calculation of the ordinates of a curve in terms of the versines.

Let us consider a circular arc AB, $2a$ long (fig. 1).

Let y_3, y_4, y_5, be the ordinates of the

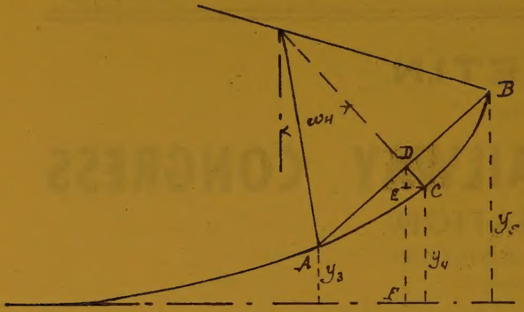

Fig. 1.

points A, C, B, and f_4 = CD, the versine at the point C, midway between A and B.

Let us then draw EC parallel to the axis of the abscissæ.

We then have

$$\mathrm{ED} = \mathrm{FD} - \mathrm{FE} = \frac{y_5 + y_3}{2} - y_4 =$$

$$\frac{y_5 - y_4}{2} - \frac{y_4 - y_3}{2} = \frac{\Delta y_5 - \Delta y_4}{2} = \frac{\Delta_2 y_5}{2}$$

$$= \mathrm{DC} \cos \omega_4 = f_4 \cos \omega_4.$$

Consequently

$$y_5 = 2\,\Sigma\Sigma_0^4 f \cos \omega$$

or in general

$$y_n = 2\,\Sigma\Sigma_0^n f \cos \omega.$$

Consequently the ordinate of a curve of any form is calculated by a double summation of the f *cos* ω. Let us then take a circular arc of a length equal to 5 equidistances a, the versines of which are respectively equal to :

$f_0, f_1, f_2, f_3, f_4, f_5$ at the pegs 0,1,2,3,4, and 5.

We wish to calculate the ordinate of peg 5 relatively to the tangent at the beginning of the curve.

All that is required is to get out the following table :

Number of the pegs. 1	f cos ω. 2	Sum of the f cos ω. 3	Sum of the numbers of column 3, $\frac{y}{2}$. 4
0	$f_0 \cos \omega_0$	$f_0 \cos \omega_0$	0
1	$f_1 \cos \omega_1$	$f_0 \cos \omega_0 + f_1 \cos \omega_1$	$f_0 \cos \omega_0$
2	$f_2 \cos \omega_2$	$f_0 \cos \omega_0 + f_1 \cos \omega_1 + f_2 \cos \omega_2$	$2f_0 \cos \omega_0 + f_1 \cos \omega_1$
3	$f_3 \cos \omega_3$	$f_0 \cos \omega_0 + f_1 \cos \omega_1 + f_2 \cos \omega_2 + f_3 \cos \omega_3$	$3f_0 \cos \omega_0 + 2f_1 \cos \omega_1 + f_2 \cos \omega_2$
4	$f_4 \cos \omega_4$	$f_0 \cos \omega_0 + f_1 \cos \omega_1 + f_2 \cos \omega_2 + f_3 \cos \omega_3 + f_4 \cos \omega_4$	$4f_0 \cos \omega_0 + 3f_1 \cos \omega_1 + 2f_2 \cos \omega_2 + f_3 \cos \omega_3$
5	$f_5 \cos \omega_5$	$f_0 \cos \omega_0 + f_1 \cos \omega_1 + f_2 \cos \omega_2 + f_3 \cos \omega_3 + f_4 \cos \omega_4 + f_5 \cos \omega_5$	$5f_0 \cos \omega_0 + 4f_1 \cos \omega_1 + 3f_2 \cos \omega_2 + 2f_3 \cos \omega_3 + f_4 \cos \omega_4$
6	$f_6 \cos \omega_6$		$6f_0 \cos \omega_0 + 5f_1 \cos \omega_1 + 4f_2 \cos \omega_2 + 3f_3 \cos \omega_3 + 2f_4 \cos \omega_4 + f_5 \cos \omega_5$

This gives

$$\frac{y_5}{2} = 5f_0 \cos \omega_0 + 4f_1 \cos \omega_1 + 3f_2 \cos \omega_2$$
$$+ 2f_3 \cos \omega_3 + f_4 \cos \omega_4$$

or, in general terms,

$$\frac{y^n}{2} = nf_0 \cos \omega_0 + (n-1)\,f_1 \cos \omega_1 +$$
$$(n-2)\,f_2 \cos \omega_2 + \ldots\ldots + f_{n-1} \cos \omega_{n-1} \quad (1)$$

We will now check this equation in the case of a circular arc.

The versine at the peg 0 (tangent point) is equal to $\frac{f}{2}$, all the other versines are equal to f, and the above equation becomes :

$$\frac{y_5}{2} = 5\frac{f}{2}\cos\omega_0 + 4f\cos\omega_1 + 3f\cos\omega_2 + 2f\cos\omega_3 + f\cos\omega_4 \quad . \; . \; . \quad (2)$$

Let us calculate the angles ω.

At peg 0, $\omega = 0$,

at peg 1, $\omega_1 = \frac{a}{R} = \frac{a \times 2f}{a^2} = \frac{2f}{a}$, since, without appreciable error, $f = \frac{a^2}{2R}$ for the equidistance a,

at peg 2, $\omega_2 = \frac{2a}{R} = \frac{2 \times 2f}{a}$

at peg 3, $\omega_3 = \frac{3a}{R} = \frac{2 \times 3f}{a}$

at peg n, $\omega_n = \frac{2 \times nf}{a}$.

Now to calculate the value of the cosines ω.

We know that

$$\cos\omega = 1 - \frac{\omega^2}{2!} + \frac{\omega^4}{4!} - \frac{\omega^6}{6!}.$$

Neglecting the terms above the second degree, which are very small, we get :

$$\cos\omega = 1 - \frac{\omega^2}{2}.$$

Substituting in this equation the value found above for ω, we obtain :

$$\left.\begin{aligned} &\cos\omega_0 = 1, \ \cos\omega_1 = 1 - \frac{1}{2}\left(\frac{2f}{a}\right)^2, \\ &\cos\omega_2 = 1 - \frac{1}{2}\left(\frac{2 \times 2f}{a}\right)^2, \\ &\cos\omega_3 = 1 - \frac{1}{2}\left(\frac{2 \times 3f}{a}\right)^2, \\ &\cos\omega_4 = 1 - \frac{1}{2}\left(\frac{2 \times 4f}{a}\right)^2, \\ &\cos\omega_n = 1 - \frac{1}{2}\left(\frac{2 \times nf}{a}\right)^2. \end{aligned}\right\} \quad (3)$$

If we replace in equation 2 the cosines ω by their values, this equation becomes :

$$\frac{y_5}{2} = \frac{5f}{2} + 4f\left[1 - \frac{1}{2}\left(\frac{2f}{a}\right)^2\right] + 3f\left[1 - \frac{1}{2}\left(\frac{2 \times 2f}{a}\right)^2\right] + 2f\left[1 - \frac{1}{2}\left(\frac{2 \times 3f}{a}\right)^2\right] + $$

$$+ f\left[1 - \frac{1}{2}\left(\frac{2 \times nf}{a}\right)^2\right]$$

$$\frac{y_5}{2} = \frac{5f}{2} + 4f + 3f + 2f + f - \frac{f^3}{2a^2}[4 \times (2 \times 1)^2 + 3 \times (2 \times 2)^2 + 2 \times (2 \times 3)^2 + 1 \times (2 \times 4)^2]$$

and, in general terms :

$$\frac{y_n}{2} = \frac{nf}{2} + (n-1)f + \ldots\ldots 2f + f - \frac{f^3}{2a^2}[(n-1)(2 \times 1)^2 + (n-2)(2 \times 2)^2 + \ldots\ldots 2(2(n-2))^2 + 1 \times (2(n-1))^2]$$

$$\frac{y_n}{2} = \frac{nf}{2} + n\frac{(n-1)}{2}f - \frac{f^3}{2a^3}[(n-1)(2 \times 1)^2 + (n-2)(2 \times 2)^2 + \ldots . 2(2(n-2))^2 + 1 \times (2(n-1))^2]$$

$$\frac{y_n}{2} = \frac{n^2 f}{2} - \frac{f^3}{2a^2}[(n-1)(2 \times 1)^2 + (n-2)(2 \times 2)^2 + \ldots\ldots 2(2(n-2))^2 + 1 \times (2(n-1))^2].$$

Applying this equation to the calculation of the ordinates y_3 of a circular arc of 100 metres radius and 30 metres developed length, we get :

$$n = 3, \qquad f = 0.4996, \qquad a = 10 \text{ m}.$$

whence :

$$y_3 = 2\left(\frac{3^2}{2} \times 0.4996\right) - \frac{2 \times \overline{0.4996}^3}{2 \times 100}\left[(3-1)(2\times 1)^2 + 1(2\times 2)^2\right] = 4.466.$$

This is the value obtained by the usual method.

Let us now consider any curve the versines of which are $f_0, f_1, f_2, f_3, f_4, f_5 \ldots$ We know that

$$\cos\omega = \frac{1}{\sqrt{1 + \mathrm{tg}^2\,\omega}}$$

We are therefore obliged to calculate the values of the tangents ω.

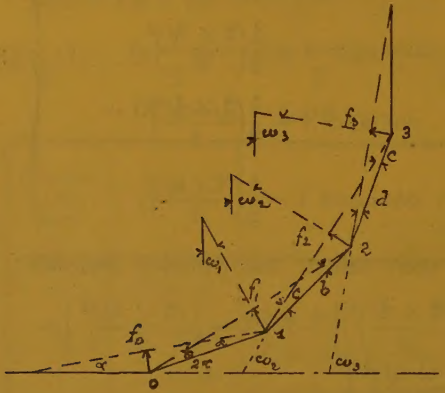

Fig. 2.

We have (fig. 2) :

$$\omega_1 = 2\alpha + b$$
$$\omega_2 = 2\alpha + b + b + c = 2\alpha + 2b + c$$
$$\omega_3 = 2\alpha + 2b + c + d + c = 2\alpha + 2b + 2c + d.$$

The angles α, b, c, d, etc., being very small, when the equidistance a is suitably chosen, the angles can be taken as equal to their tangents and the sum of the angles be taken as equal to the sum of the tangents.

Consequently, noting that :

$$\tan\alpha = \frac{f_0}{a}, \ \tan b = \frac{f_1}{a}, \ \tan c = \frac{f_2}{a}, \ \tan d = \frac{f_3}{a},$$

we find

$$\tan\omega_1 = \frac{2f_0}{a} + \frac{f_1}{a} = \frac{2f_0 + f_1}{a}$$

$$\tan\omega_2 = \frac{2f_0}{a} + \frac{2f_1}{a} + \frac{f_2}{a} = \frac{2f_0 + 2f_1 + f_2}{a}$$

$$\tan\omega_3 = \frac{2f_0}{a} + \frac{2f_1}{a} + \frac{2f_2}{a} + \frac{f_3}{a}$$
$$= \frac{2f_0 + 2f_1 + 2f_2 + f_3}{a}.$$

and in general

$$\tan\omega_n = \frac{2f_0 + 2f_1 + 2f_2 + \ldots\ldots + f_n}{a} \qquad (4)$$

and

$$\cos\omega_n = \frac{1}{\sqrt{1 + \left(\frac{2f_0 + 2f_1 + 2f_2 + \ldots f_n}{a}\right)^2}} \qquad (5)$$

The cosines ω being calculated, equation (1) enables us to calculate with sufficient exactitude, the ordinate at any point of a curve the versines of which are known.

Now to check this equation by applying it to the calculation of the ordinate of a circular arc of 1 000 metres radius and 200 metres developed length, we find, taking an equidistance equal to 10 m. $y_{20} = 19.936$ metres, the value given by the geometric method.

In the same way, we should find that

the ordinate of a circular arc of 500 metres radius and 100 metres developed length is :

$y_{10} = 9.967$, again the value given by the tables of ordinates. And for the ordinate of a circular arc of 100 metres radius and 30 metres developed length : $y_3 = 4.466$, value given by the tables of ordinates and found previously.

It would also be quite easy to check that the equations (3) and (5) give equal numerical values for the cosines ω.

Exact relations between the versines of two curves of any form having the same ends.

As we know how to calculate the ordinates of a curve in terms of the versines, let us now consider two curves of any form, C and C_1 (fig. 3) sufficiently close together and with very slightly different developed lengths, so that the equidistant pegs of the same row starting from the point at which chaining begins, O, such as A and D, B and E, C and F, can be taken as being placed on the same ordinate.

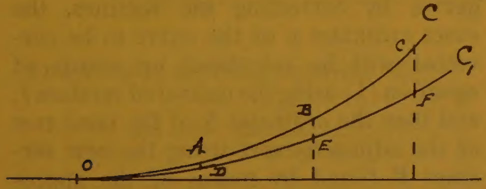

Fig. 3.

C is the curve to be corrected and its versines are those marked *f*.

C_1 is the corrected curve and the new versines are marked F.

Let us suppose that these two curves are joined together at their end at the pegs 4 and 5. Aplying equation (1) we get :

— at peg 5 of curve C :

$$\frac{y_5}{2} = 5f_0 \cos \omega_0 + 4f_1 \cos \omega_1 + 3f_2 \cos \omega_2 + 2f_3 \cos \omega_3 + f_4 \cos \omega_4$$

— at the same peg of curve C_1 :

$$\frac{Y_5}{2} = 5F_0 \cos \Omega_0 + 4F_1 \cos \Omega_1 + 3F_2 \cos \Omega_2 + 2F_3 \cos \Omega_3 + F_4 \cos \Omega_4$$

— at peg 4 of curve C :

$$\frac{y_4}{2} = 4f_0 \cos \omega_0 + 3f_1 \cos \omega_1 + 2f_2 \cos \omega_2 + f_3 \cos \omega_3$$

— at the same peg of curve C_1 :

$$\frac{Y_4}{2} = 4F_0 \cos \Omega_0 + 3F_1 \cos \Omega_1 + 2F_2 \cos \Omega_2 + F_3 \cos \Omega_3.$$

We ought, therefore, to have : $y_5 = Y_5$, that is to say :

$$5f_0 \cos \omega_0 + 4f_1 \cos \omega_1 + 3f_2 \cos \omega_2 + 2f_3 \cos \omega_3 + f_4 \cos \omega_4 = 5F_0 \cos \Omega_0 + 4F_1 \cos \Omega_1 + 3F_2 \cos \Omega_2 + 2F_3 \cos \Omega_3 + F_4 \cos \Omega_4,$$

or again :

$$\frac{y_5}{2} - \frac{Y_5}{2} = 5f_0 \cos \omega_0 - 5F_0 \cos \Omega_0 + 4f_1 \cos \omega_1 - 4F_1 \cos \Omega_1 + 3f_2 \cos \omega_2 - 3F_2 \cos \Omega_2 + 2f_3 \cos \omega_3 - 2F_3 \cos \Omega_3 + f_4 \cos \omega_4 - F_4 \cos \Omega_4 = 0$$

and

$$\frac{y_4}{2} - \frac{Y_4}{2} = 4f_0 \cos \omega_0 - 4F_0 \cos \Omega_0 + 3f_1 \cos \omega_1 - 3F_1 \cos \Omega_1 + 2f_2 \cos \omega_2 - 2F_2 \cos \Omega_2 + f_3 \cos \omega_3 - F_3 \cos \Omega_3 = 0,$$

or, in general terms :

$$nf_0 \cos \omega_0 - nF_0 \cos \Omega_0 + (n-1)\, f_1 \cos \omega_1 - (n-1)\, F_1 \cos \Omega_1 \ldots + f_{n-1} \cos \omega_{n-1} - F_{n-1} \cos \Omega_{n-1} = 0$$

Such is the exact relation between the versines f and F of curves of any form, lying near together and connecting two straight sections.

The angles ω and Ω being as a rule very small, by making $\cos\omega = \cos\Omega = 1$, we get nearly equal values for the ordinates y_5, Y_5, y_4, Y_4. Calling them a_5, A_5, a_4, A_4, we can write :

$$\frac{a_5}{2} = 5f_0 + 4f_1 + 3f_2 + 2f_3 + f_4 = \frac{y_5}{2} - e_5$$

$$\frac{A_5}{2} = 5F_0 + 4F_1 + 3F_2 + 2F_3 + F_4 = \frac{Y_5}{2} - E_5.$$

$$\frac{a_4}{2} = 4f_0 + 3f_1 + 2f_2 + f_3 = \frac{y_4}{2} - e_4$$

$$\frac{A_4}{2} = 4F_0 + 3F_1 + 2F_2 + F_3 = \frac{Y_4}{2} - E_4.$$

By subtraction, term by term, we find :

$$\frac{a_5 - A_5}{2} = \frac{y_5 - Y_5}{2} + E_5 - e_5 =$$
$$5(f_0 - F_0) + 4(f_1 - F_1) + 3(f_2 - F_2) + 2(f_3 - F_3) + (f_n - F_n)$$

whence, making $\delta = (f - F)$

$$5\delta_0 + 4\delta_1 + 3\delta_2 + 2\delta_3 + \delta_4 = \Sigma_0^5 \mathfrak{M}'\delta = E_5 - e_5 \quad . \quad . \quad . \quad . \quad (6)$$

since

$$\frac{y_5 - Y_5}{2} = 0;$$

$$\frac{a_4 - A_4}{2} = \frac{y_4 - Y_4}{2} + E_4 - e_4 =$$
$$4(f_0 - F_0) + 3(f_1 - F_1) + 2(f_2 - F_2) + (f_3 - F_3)$$

whence

$$4\delta_0 + 3\delta_1 + 2\delta_2 + \delta_3 = \Sigma_4^0 \mathfrak{M}'\delta = E_4 - e_4 \quad . \quad . \quad . \quad . \quad (7)$$

since

$$\frac{y_4 - Y_4}{2} = 0.$$

Subtracting, term by term, in equations (6) and (7), we find :

$$\delta_0 + \delta_1 + \delta_2 + \delta_3 + \delta_4 = (E_5 - E_4) - (e_5 - e_4)$$

or, in general terms :

$$n\delta_0 + (n-1)\delta_1 + \ldots \delta_{n-1} = \Sigma_0^n \mathfrak{M}'\delta = E_n - e_n$$

$$\delta_0 + \delta_1 + \delta_2 + \delta_3 + \delta_n \ldots\ldots \delta_n = \Sigma_0^n \delta$$
$$= (E_n - E_{n-1}) - (e_n - e_{n-1}).$$

Consequently for two curves lying close together and of any form, connecting two straight sections, the sum of the differences between the versines of the same row of pegs, as that of the sum of the moments of these differences are not nil.

However, if we consider two adjacent curves, the errors E_n and e_n, E_{n-1} and e_{n-1} have practically equal values since these are the errors committed when calculating the ordinates on the same row and of little different values; it follows that the difference $E_n - e_n$ and $[(E_n - E_{n-1}) - (e_n - e_{n-1})]$ are very close to 0 as we shall see in the following examples.

Consequently, to ascertain the degree of accuracy of the method of adjusting curves by correcting the versines, the exact ordinates y of the curve to be corrected will be calculated by means of equation (1) using the measured versines f, and then the ordinates Y of the same row of the adjusted curve using the new versines F found by means of the approximate equations

$$\Sigma(f - F) = 0 \text{ and } \Sigma\mathfrak{M}'(f - F) = 0.$$

The differences $y - Y$ obviously measure, at each peg, the distances between the two curves.

The amounts separating these differences and the movements required d calculated by the equation $\Sigma\mathfrak{M}'(f - F)$ give the error committed at each peg and consequently the degree of accuracy of the method.

Error committed when using, for the purpose of adjusting a curve, the approximations

$$\Sigma(f - F) = 0 \qquad \Sigma \mathcal{M}^t (f - F) = 0.$$

In order to make the point clear, a number of numerical examples will now be dealt with.

a) We will first of all consider a deformed parabolic transition the versines of which in millimetres (mm.) are equal to :

$f_0 = 8, f_1 = 10, f_2 = 16, f_3 = 27, f_4 = 36,$
$f_5 = 64, f_6 = 46.$

Let us make $F_0 = 5$, $F_1 = 11$, $F_2 = 21$, $F_3 = 29$, $F_4 = 38$, $F_5 = 47$, $F_6 = 56$, the new versines of the adjusted transition, calculated by using the known fundamental equations

$$\Sigma(f - F) = 0 \qquad \Sigma \mathcal{M}_t (f - F) = 0.$$

The amounts the deformed curve has to be moved to get it to the correct position are as much as 20 mm.

The application of equation (4) gives for the deformed curve :

$$\tan \omega_1 = \frac{2 \times 8 + 10}{10} = 2.6 \text{ mm.}$$

$$\tan \omega_2 = \frac{2(8 + 10) + 16}{10} = 5.2 \text{ mm.}$$

$$\tan \omega_3 = \frac{2(8 + 10 + 16) + 27}{10} = 9.5 \text{ mm.}$$

$$\tan \omega_4 = \frac{2(8 + 10 + 16 + 27) + 36}{10} = 15.8 \text{ mm.}$$

$$\tan \omega_5 = \frac{2(8 + 10 + 16 + 27 + 36) + 64}{10} = 25.8 \text{ mm.}$$

$$\tan \omega_6 = \frac{2(8 + 10 + 16 + 27 + 36 + 64) + 46}{10} = 36.8 \text{ mm.}$$

And finally, applying equation (5):

$$\cos \omega = \frac{1}{\sqrt{1 + \mathrm{tg}^2 \, \omega}}$$

we get

$$\cos \omega_1 = \cos \omega_2 = \cos \omega_3 = 1,$$
$$\cos \omega_4 = 0.9998, \ \cos \omega_5 = 0.9997,$$
$$\cos \omega_6 = 0.9994.$$

The same calculations for the adjusted curve give :

$$\cos \Omega_1 = \cos \Omega_2 = \cos \Omega_3 = \cos \Omega_4 = 1,$$
$$\cos \Omega_5 = 0.9997, \ \cos \Omega_6 = 0.9994.$$

Let us calculate the ordinate $\frac{y_6}{2}$ of the curve to be adjusted, using equation (1). We have

$$\frac{y_6}{2} = 6 \times 8 \times 1 + 5 \times 10 \times 1 + 4 \times 16 \times 1$$
$$+ 3 \times 27 \times 0.9998 + 2 \times 36 \times 0.9997$$
$$+ 64 \times 0.9994 = 378.96 \text{ mm.}$$

We also get :

$$\frac{Y_6}{2} = 6 \times 5 \times 1 + 5 \times 11 \times 1 + 3 \times 29 \times 1$$
$$+ 2 \times 38 \times 0.9997 + 1 \times 47 \times 0.9994$$
$$= 378.97 \text{ mm.}$$

Consequently

$$\frac{y_6}{2} - \frac{Y_6}{2} = \frac{1}{100} \text{ mm.}$$

so that the two curves pass through peg 6.

We should also find that :

$$\frac{y_7}{2} = 0.58593 \text{ and } \frac{Y_7}{2} = 0.58594,$$

so that the two curves pass through pegs 6 and 7.

In addition, it is easy to verify that the amounts of the movements d calculated with equation $\Sigma \mathcal{M}^t (f - F)$ equal the differences $y - Y$ between the ordinates.

In the case, quite usual, of these parabolic transitions, the fundamental formulæ

$$\Sigma (f - F) = 0 \text{ and } \Sigma \mathcal{M}^t (f - F) = 0$$

are therefore very accurate.

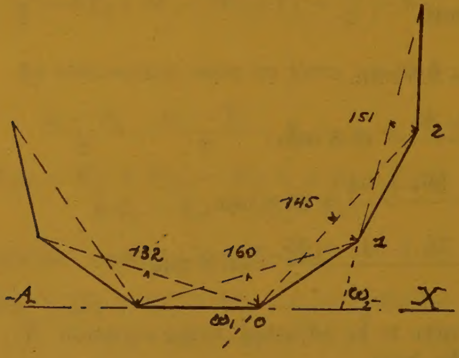

Fig. 4.

b) Let us consider, figure 4, the part of the deformed curve the versines of which are, in millimetres :

$f_0 = 160$, $f_1 = 145$, $f_2 = 151$, $f_3 = 135$, $f_4 = 152$, $f_5 = 156$, $f_6 = 112$, $f_7 = 95$, $f_8 = 133$, $f_9 = 185$, $f_{10} = 168$, $f_{11} = 140$, $f_{12} = 151$;

So that, in millimetres :

$F_0 = 143$, $F_1 = 144$, $F_2 = 144$, $F_3 = 144$, $F_4 = 144$, $F_5 = 144$, $F_6 = 145$, $F_7 = 146$, $F_8 = 146$, $F_9 = 146$, $F_{10} = 147$, $F_{11} = 146$, $F_{12} = 144$, the versines of the adjusted curve, calculated from the approximate fundamental formulæ recalled in the preceding example.

The amounts the deformed curve has to be moved to bring it to its correct position are as much as 0.30 metre.

Applying equation (4), giving the values of the tangents ω and tangents Ω, equation (5) giving the values of cosines ω and cosines Ω, and equation (1) giving the values of the ordinates, we get :

$$\frac{y_{12}}{2} = 11.1466 \qquad \frac{Y_{12}}{2} = 11.144$$

$$\frac{y_{13}}{2} = 12.9887 \text{ and } \frac{Y_{13}}{2} = 12.9855\ (^1).$$

The new adjusted curve as determined by the new versines F, passes therefore at 2 (11.1466 — 11.144) = 2.6 mm. × 2, from peg 12 of the former curve and at 2 (12.9887 — 12.9855) = 3.2 mm. × 2 from peg 13 of the former curve.

For this curve to pass through pegs 12 and 13, it is easy to see that it would be sufficient to alter the versine F_0 by

$$\frac{3.2 \text{ mm.}}{13} = 2/10 \text{ mm. approximately.}$$

In addition, if we observe that $(Y_{13} - y_{13}) - (Y_{12} - y_{12}) = 0.0006$, we can conclude that practically

$$\Sigma (f - F) = 0.$$

In the same way, we can verify that the displacements d calculated from the equation $\Sigma \mathcal{M}^t (f - F)$ differ but little from the differences $(y - Y)$ and that these discrepancies can be cancelled out by a modification of the versines F of the order of one tenth of a millimetre.

Consequently, although the radius of the curve to be adjusted is relatively small (about 350 m.) and the displacements of the pegs of the deformed curve are as much as 0.30 m., the fundamental formulæ

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(1) The ordinates are measured relatively to the chord AX.

$$\Sigma(f-\mathrm{F})=0 \quad \text{and} \quad \Sigma\,\mathcal{M}^t(f-\mathrm{F})=0$$

still give quite acceptable results.

Conclusions. — The examples of the adjustment of deformed curves we have dealt with show that the use of the equations

$$\Sigma(f-\mathrm{F})=0 \quad \text{and} \quad \Sigma\,\mathcal{M}^t(f-\mathrm{F})=0,$$

for ascertaining the values of the new versines and the calculation of the displacements, gives very exact results.

Our demonstration has been based on the condition of comparing adjacent curves, the usual problem to solve, but the theory can be made of general application.

When comparing two widely separated curves, the first investigation into the adjustment will give a slightly irregular layout, the irregularities of which will disappear after a second correction, because we shall then be dealing with curves lying very close together.

If we realise that the measurements to be made on the ground are very easily done with accuracy, we can conclude that this method is much to be preferred to the classical field method.

* * *

Rule to ascertain if a circular arc was correctly laid in originally.

Take a deformed circular arc correctly laid in originally.

Let F be the versine of this circular arc. When a peg n is displaced towards the outside of the curve by d millimetres, the versine F at this peg becomes F + d and the versine at the pegs $n-1$ and $n+1$ are reduced by

$$\delta=\frac{d}{2}\cos\omega, \text{ figure 5.}$$

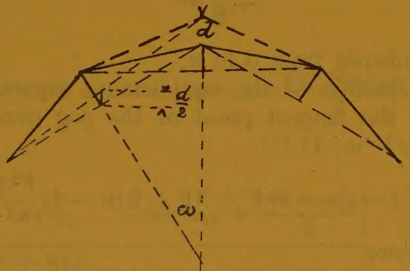

Fig. 5.

(If this peg was displaced towards the inside of the curve, its versine F would be reduced by d, and the two adjacent versines increased by δ).

But $\cos\omega=1-\frac{\omega^2}{2}$ without appreciable error, and $\omega=\frac{a}{\mathrm{R}}$, a equidistance between the pegs, whence :

$$\delta=\frac{d}{2}\left(1-\frac{a^2}{2\mathrm{R}^2}\right)=\frac{d}{2}-\frac{d\,a^2}{4\mathrm{R}^2}=\frac{d}{2}-\frac{\mathrm{F}^2 d}{a^2}$$

when noting that $\mathrm{F}=\frac{a^2}{2\mathrm{R}}$,

and we have :

$$f_{n-1}=\mathrm{F}-\frac{d}{2}+\frac{\mathrm{F}^2 d}{a^2}$$

$$f_n=\mathrm{F}+d$$

$$f_{n+1}=\mathrm{F}-\frac{d}{2}+\frac{\mathrm{F}^2 d}{a^2}$$

If we agree that all the pegs but the first and the last (which are located on the straight sections) are moved by $+\,d$ millimetres, we find, making the sum of the versines of the deformed curve, and the sum of the versines of the theoretical curve equal :

$$\frac{F}{2} + F + F + \ldots + F + \frac{F}{2}\left[(N+1)^2 - 2N^2\right] + \frac{F}{2}N^2$$

numbered from 0 to n,
N, fraction of the equidistance separating the tangent point of the peg numbered $(n-1)$ (¹).

$$\Sigma_0^n f = (n-1)\,F + NF + 2(n-1)\frac{F^2 d}{a^2}$$

whence

$$F = \frac{\Sigma_0^n f}{n-1+N} - \frac{2(n-1)}{n-1+N}\,\frac{F^2 d}{a^2}$$

When $F = 0.10$, $(R = 500)$, $d = 0.20$, $a = 10$ m., $N = 0.1$, the second factor of the second member is equal to 0.00004.

Generally the pegs are not all moved in the same direction and by the same quantity d, and in this case, the value of this second term is still smaller so that, without appreciable error :

$$F = \frac{\Sigma_0^n f}{n-1+N}$$

A deformed circular arc will therefore have been properly laid in originally when the quotient of the sum of the measured versines by the term $n - 1 + N$ *is equal to the theoretical versine of the circular arc.*

When the circular arc is connected to the straight sections by parabolic transitions, and the curve was properly laid in originally, the versine of the original circular arc ought to be found by dividing the sum of the versines measured on the deformed layout having parabolic transitions by the term $n - 1 + N$ corresponding to the original circular arc, since the sum of the versines of this latter is equal to the sum of the versines of the deformed layout with parabolic transitions.

The checking of a curve formed by two parabolic transitions can, however, be done directly as we shall see below.

* * *

Rule for ascertaining if a curve formed of two parabolic transitions was properly laid in originally.

a) *Curve formed of two parabolic transitions separated by a circular arc of a length equal to one equidistance.*

Let us suppose that each parabolic transition spread over 10 equidistances is 100 m. long, and that K is the difference between two succeeding versines F.

The versines of the curve are equal to :

$$\frac{K}{6}, K, 2K, 3K, 4K, 5K, 6K, 7K, 8K, 9K, 10K - \frac{K}{6}.$$

$$10K - \frac{K}{6},\ 9K,\ 8K,\ \ldots\ldots\ K,\ \frac{K}{6}.$$

As we have seen above, when a peg n is displaced d millimetres towards the outside of the curve, the versine at this peg becomes $f = F + d$, and the versine at the pegs $n - 1$ and $n + 1$ are reduced by

$$\delta = \frac{d}{2}\cos\omega$$

(If this peg were moved towards the inside of the curve, its versine F would be reduced by d and the two adjacent versines increased by δ).

If we agree that all the pegs, except

(¹) See : Diagrams of versines and superelevations (J. Chappellet), *Bulletin of the International Railway Congress Association*, October 1930.

the end ones placed in the straight section, are moved $+ d$ millimetres we find :

$$\Sigma_0^{20+1} f = 2\frac{(1+10)}{2} 10K + 2 \times 10 d - d\left(\frac{\cos\omega_1}{2} + \cos\omega_2 + \cos\omega_3 + \ldots + \cos\omega_{20} + \frac{\cos\omega_{20+1}}{2}\right)$$

ω_1, ω_2, etc., being the angles between the perpendiculars to the correct transition at each peg.

In general, we would have :

$$\Sigma_0^{2n+1} f (1+n) n K + 2nd - d\left(\frac{\cos\omega_1}{2} + \cos\omega_2 + \cos\omega_3 \ldots + \cos\omega_{2n} + \frac{\cos\omega_{2n+1}}{2}\right)$$

and

$$K = \frac{\Sigma_0^{2n+1} f}{(1+n)n} - \frac{2nd + d\left(\frac{\cos\omega_1}{2} + \cos\omega_2 + \cos\omega_3 \ldots + \cos\omega_{2n} + \frac{\cos\omega_{2n+1}}{2}\right)}{(1+n)n} \qquad (8)$$

Now to calculate $\cos\omega_1$.

Without appreciable error,

$$\omega_1 = \frac{a}{\rho_1},$$

ρ_1 radius of the osculatory circle at peg 1 of the transition and a equidistance between the pegs.

We showed (see *Bulletin of the International Railway Congress Association*, October 1930, fig. 25, page 2016) that, without appreciable error, for a given chord, the versine at a point of a parabolic transition is equal to that of the circle osculatory to it at this point.

Neglecting the terms above the second degree, we have :

$$\cos\omega_1 = 1 - \frac{\omega^2_1}{2} = 1 - \frac{a^2}{2\rho^2_1}$$

but, without appreciable error,

$$F_1 = \frac{a^2}{2\rho_1}$$

F_1, versine of the parabolic transition at peg 1, a : equidistance between the pegs, whence

$$\rho_1 = \frac{a^2}{2F_1}, \quad \rho_1^2 = \frac{a^4}{4F_1^2}$$

and

$$\cos\omega_1 = 1 - \frac{a^2}{\frac{2a^4}{4F^2_1}} = 1 - \frac{2F_1^2}{a^2},$$

We would also find :

$$\cos\omega_n = 1 - \frac{2F^2_n}{a^2}.$$

As we know how to calculate the value of all the cosines ω, we could deduce K therefrom, but the calculations would be laborious; to make them easier, we will verify that the numerical value of the second term of the second number of formula (8) may be overlooked and that we can write :

$$K = \frac{\Sigma_0^{2n+1} f}{(1+n)n}.$$

For this purpose, let us suppose that the two parabolic transitions are 100 m. long and are osculatory to a circular arc of 500 metres radius, the versine of which, for $a = 10$ metres, is equal to 0.10 metre. We get $K = \frac{0.10}{10} = 0.01$ and $n = 10$. Making $d = 0.10$ we find that the second term of the second member of formula (8) is equal to 0.00000132, that is to say it is practically nil and may be overlooked.

The pegs are generally not all displaced in the same direction and to the same extent d, and in this case the error committed is still smaller, and we have still, without appreciable error :

$$K = \frac{\Sigma_0^{2n+1} f}{(1+n)n}.$$

A deformed curve composed of two parabolic transitions, separated by a circular arc one equidistance in length, will therefore have been well laid in originally if the quotient of the sum of the measured versines by the product $(1+n)\,n$ *is equal to the difference* K *between two succeeding versines of the theoretical parabolic transition.*

b) *The curve is composed of two parabolic transitions placed end to end.*

The versines of the curve are respectively equal to :

$$\frac{K}{6}, K, 2K, 3K, 4K, 5K, 6K, \ldots.. 9K, 10K - \frac{K}{3},$$

$$9K, 8K, 7K, \ldots.. K, \frac{K}{6}.$$

Under the same conditions as before, we have :

$$K = \frac{\Sigma_0^{2n} f}{n^2}.$$

Consequently, a curve composed of two parabolic transitions placed end to end will have been laid in properly originally if the quotient of the sum of the measured versines of the curve deformed by n^2 *is equal to the difference* K *between two successive versines of the theoretical parabolic transition.*

We have supposed that the parabolic transitions have a developed length which is a multiple of the equidistance; in the contrary case, the formula giving the coefficient K would be arrived at by using as the values of the versines those obtained from the formulæ given on page 2021 to 2025 of the October, 1930, *Bulletin of the International Railway Congress Association.*

The rules given above, by which F and K can be calculated can be very useful when carefully investigating the adjustment of a deformed curve. In fact when the latter was originally properly laid in, the displacements needed to replace it in its correct position are generally small and always easily carried out, provided no structure or other obstacle has been built on or near this badly traced curve.

[625. 113]

Progressive transition curves on railways,

by Carlos LAFFITTE MARTINEZ,

Industrial Engineer,

The *Bulletin of the International Railway Congress Association* for December, 1932, reproduced the report by Professor H. Baumann, Direktor bei der Deutschen Reichsbahn (German State Railways), and Mr. Jaehn, Reichsbahnoberrat (of the same Railways), dealing with the « Relationship between the vehicle and the track to ensure safety at high speeds. » When dealing with the question of joining up straight sections to circular curves, the said reporters suggested as a measure of the quality of such transition curves, the derivate in terms of the time of the acceleration resulting from the movement of the axis of the vehicle.

In an article under the same title as the present note, published in the *Revista* de *Obras Publicas* of the 15th March, 1926, we suggested a criterion of the same kind, although we took into consideration the variations in the angular acceleration instead of considering those of the acceleration of the axis of the vehicle. Subsequently we published in the *Revista de Ingenieria Industrial* (1) an article in which we endeavoured to relate the degree of « comfort » of a vehicle to the changes of acceleration the traveller experiences while the train is running.

We now propose to make some observations and comments upon the subject of the ratio mentioned at the beginning of this note; so that our ideas on the matter may be clear, we will start briefly with the part relating to the progressive transitions.

If we consider an axle of a vehicle running through a circular curve of R metres radius at the speed of V kilometres an hour, the superelevation of the curve being h millimetres, the centrifugal acceleration in metres per second will be $\frac{V^2}{12.96\,R}$; the centripetal acceleration due to the superelevation and expressed in the same unit, supposing the gauge to be 1.50 metres between the centres of rails, is $\frac{h}{153}$, and the resultant — which the reporters state is adequate — will be expressed by

$$p \sim \frac{V^2}{13\,R} - \frac{h}{153} \quad . \quad . \quad . \quad (1)$$

To calculate the change of acceleration, the authors of the report in question use the formula $\psi = \frac{p}{t}$ in which $t = \frac{l}{V}$ is the time required to run over the transition curve of length l at the speed V; they thus obtain :

$$\psi = \left(\frac{V^2}{13\,R} - \frac{h}{153}\right)\frac{V}{3.6\,l} \quad . \quad . \quad (2)$$

In the case of a vehicle of a total wheel base a, the inscription of the vehicle takes place on the distance $l + a$, and the change of acceleration becomes :

$$\psi = \left(\frac{V^2}{13\,R} - \frac{h}{153}\right)\frac{V}{3.6\,(l + a)} \quad . \quad (3)$$

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(1) See the July, 1930, number of this review.

In the case in which there is no transition curve and no superelevation: $h = 0$, and $l = 0$ and

$$\psi = \frac{V^3}{46.7 Ra} \quad . \quad . \quad . \quad (4)$$

Observations.

The calculation of ψ by means of equation (3) gives approximately the average change of acceleration when passing from the straight on to the curve; in our opinion, however, the strict interpretation of the value of the change of acceleration, the mathematical expression for which is $\psi = \frac{dp}{dt}$, does not lead to the said formula. Actually, the passage of the vehicle from the straight to the circular curve can be decomposed into three distinct movements, namely :

1. entry of the vehicle onto the transition, that is to say with one pair of wheels or one bogie on the straight and the other on the transition (¹);

2. running through the transition — the two bogies are on the transition;

3. entry on to the circular curve — one bogie is on the transition and the other on the circular curve.

Let us take the second case, and suppose that the length of the superelevation ramp coincides with that of the transition. The radius of curvature of the path described by the axis of the vehicle represents the half sum of the radii of the paths described by the bogie axles, and in view of the properties of the cubic parabola, we get approximately for this radius :

$$\frac{1}{\rho} = \frac{x}{Rl} \quad . \quad . \quad . \quad . \quad (5)$$

x being the distance from the axis of the vehicle to the origin of the transition.

The amount the vehicle is raised in the position considered is $h\frac{x}{l}$, and the resulting acceleration :

$$p = \frac{V^2 x}{13 Rl} - h\frac{x}{153 l}$$

Making $x = Vt$ and substituting the value of t, we get

$$\psi = \frac{dp}{dt} = \left(\frac{V^2}{13R} - \frac{h}{153}\right)\frac{V}{3.6 l} \qquad (6)$$

an expression identical with equation (2) of the reporters, Messrs. BAUMANN and JAEHN, with the difference that these Gentlemen established their equation for an individual axle, whereas ours is for the complete vehicle.

If we deal in the same way with case No. 1, basing our investigation upon the fact that the values of the ordinates of the curve described by the axis of the vehicle equal half the corresponding ones of case 2, and that the superelevations are also reduced by one half, we obtain

$$\psi = \left(\frac{V^2}{13 Rl} - \frac{h}{153}\right)\frac{V}{2 \times 3.6 l} \quad . \quad (7)$$

In case 3, we should find the same value (7) for the change of acceleration; in other terms the maximum or most unfavourable value corresponds to case 2 and is given by equation (6), which, as we see, gives values independent of a.

Considering the case without transition curve, we get for ψ the value ∞ and not that obtained in equation (4) by the reporters. This last equation gives an aver-

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(¹) Seeing how close together the bogie wheels are, we will suppose them as being a single pair in line with the bogie centre.

age value of the change of acceleration which in this case (passing direct from the straight onto the circular curve) changes suddenly from zero to $\frac{V^2}{2R}$ and from $\frac{V^2}{2R}$ to $\frac{V^2}{R}$ instead of varying gradually as where there is a progressive transition. The variation $\frac{V^2}{2R}$, in our opinion, gives in this case the mean measure of the « comfort » at the centre line of the vehicle. It will be seen without difficulty that the curve described by the said axis, one bogie being on the straight and the other on the circular curve, represents approximately a circumference of radius 2 R.

We ought also to point out that the said reporters took into consideration the centre line of the vehicle alone, and we think that from the point of view of smooth riding it is essential to consider what occurs in the part of the vehicle occupied by the passengers. The exclusive application of equation (6) or equation (3) given by the reporters does not give us any factor of comparison when the theoretical superelevation is adopted, because we then have $p = 0$ and $\psi = 0$ and this also incites us to investigate what takes place outside the centre line of the vehicle.

The calculation of the change of acceleration at each of its points is easily made in terms of the angular speed, a simple calculation which we gave in the above mentioned article in the *Revista de Obras Publicas*. The values obtained are :

for cases 1 and 3 :

$$\frac{d\omega}{dt} = \frac{V^2 x}{R l a} \quad . \quad . \quad . \quad . \qquad (8)$$

and for case 2 :

$$\frac{d\omega}{dt} = \frac{V^2}{R l} \quad . \quad . \quad . \quad . \qquad (9)$$

We see at once that, in case 2, the change of acceleration is the same for the whole of the vehicle; it can be calculated therefore by equation (6). This is not so, however, in cases 1 and 3. Let us consider, for example, the leading bogie centre of the vehicle; the acceleration will be :

$$\frac{V^2 x}{26 R l} + \frac{V^2 x}{13 R l a} \times \frac{a}{2} - \frac{h x + \frac{a}{2}}{306\, l}$$

and

$$\psi = \left(\frac{V^2}{13 R} - \frac{h}{306}\right) \frac{V}{3.6 l} \quad . \quad . \qquad (10$$

For the trailing centre we get

$$\psi = \frac{-\, h V}{306 \times 3.6 l} \quad . \quad . \qquad (11)$$

The changes of acceleration at the other points of the vehicle, and in particular at the ends where the values are maximum, can be calculated in the same way. In our opinion, in view of the layout of railway vehicles, the values at the bogie centres generally near the ends occupied by passengers ought to be considered as being unfavourable.

The changes of acceleration at the bogie centres can be calculated very easily.

We, therefore, recommend that equations (10) and (11) be used as a measure of the quality of transition curves from the point of view of smooth riding, as these give — particularly (10) — higher values of ψ than those obtained from equation (7).

These equations enable us, as we have already said, to satisfy ourselves as to the quality of a transition of given length, and also to arrive at the length of transition required for the change of acceleration to have a specified value. Messrs. Baumann and Jaehn suggest that the length of the superelevation ramp, equal to that of the transition, should be so fixed that the slope of the first be $\leqslant \frac{1}{10\,V}$ and this condition ensures lengths of transitions giving moderate changes of acceleration.

In practice, especially on secondary lines, the length of the superelevation ramp frequently differs from that of the transition : let us study this case, supposing that the first be $L \geqslant 1 + 2a$ (a = distance between bogie centres), the difference $L - l$ being divided between the ends of the progressive transition; under these conditions the centripetal acceleration in cases 1 and 3 is $\frac{h\,(x + d)}{153\,L}$, in which d represents the distance between the origin of the transition and that of the superelevation ramp. Formerly we found in these cases the fraction :

$$\frac{h\left(x + \frac{a}{2}\right)}{306\,l}$$

a result due to one bogie alone being, for $L = l$, on the superelevation ramp, whereas now both are on it, at the beginning and at the end of the transition.

This increase of the centripetal acceleration gives rise to the following change of acceleration :

$$\psi = \left(\frac{V^2}{13\,R\,l} - \frac{h}{153\,L}\right)\frac{V}{3\ 6} \qquad (12)$$

a smaller value than that given for the same case by equation (10), so that this solution allows ψ to be reduced for a calculated value of l, or l to be reduced whilst maintaining the value of ψ constant. This is of great importance, in particular when it is a question of introducing or increasing the transitions on existing lines.

In this case the change of acceleration at the axis of the vehicle must be checked to see if it is really suitable . Its value is given by :

$$\psi = \left(\frac{V^2}{26\,R\,l} - \frac{h}{153\,L}\right)\frac{V}{3.6} \quad . \quad (13)$$

At the same time care must be taken to make sure that the value of ψ is not excessive before the bogie runs onto the transition curve by using the following equation :

$$\psi = \frac{-\,hV}{153 \times 3.6\,L} \quad . \quad . \quad (17)$$

* ◆ *

We will now give some numerical applications, starting from the « comfortable » speed (giving smooth riding), accepted by the reporters Messrs. BAUMANN and JAEHN, corresponding to a resultant acceleration of 0.40 on the circular curve. Using this value in equation (1) we get :

$$h = 11.8\,\frac{V^2}{R} - 60 \quad . \quad . \quad (18)$$

Making h, for example, the maximum = 150 millimetres, $\frac{V^2}{R} = 17.8$.

Taking the slope of the superelevation ramp selected by Messrs. BAUMANN and JAEHN , *i. e.* 1/10 V, which for $h = 150$ gives $L = 1.5\ V$, and taking $l = L$, we find as maximum change of acceleration at the axis of the vehicle, by equation (6), $\psi = 0.0714$; at the leading bogie centre, by equation (10), $\psi = 0.161$ and at the trailing bogie centre, by equation (11), $\psi = 0.091$. The maximum value is therefore found at the leading bogie centre.

Proceeding in the same way for different values of h, we obtain values of ψ varying from 0.10 to 0.21 at the leading bogie centre of the vehicle.

Let us now see what occurs in the case

of $L \geqslant l + 2a$, and for this, let us adopt a maximum value of 0.20 for the change of acceleration, since we have obtained this figure in the previous case, and experience also shows it is quite a possible value in practice (¹).

Let us take, for example, the concrete case wherein $R = 800$ and $h = 150$, which corresponds (18) to $V = 120$. The solution $L = l$ gives $L = l = 180$ metres, a very satisfactory length of transition, but difficult to use in practice in many cases. If the value $\psi = 0.20$ be introduced in equation (12), we get for $L = 180$, $l = 120$, which lets us solve the problem with a much shorter transition than that given by the preceding equation, without departing at any moment from the value $\psi = 0.20$ at any point between the bogie centres, as can be checked by means of equations (13) and (17).

The length $L - l$ is split up between the ends of the progressive transition, the best division being to leave only a length $= a$, that is to say the distance between the bogie axles of the vehicle, to the straight, and the remainder to the circular curve.

The reason for this division is that if we lengthen the superelevation ramp on the straight excessively, we set up a centripetal acceleration which, when the vehicle runs on to the transition, changes direction and sets up unpleasant oscillations.

Messrs. **Baumann** and **Jaehn** are of the opinion that the origin of the superelevation ramp ought to coincide with the tangent point of the transition curve; we think that, seeing the advantages we have mentioned, it is desirable most often to prolong the ramp into the straight as indicated, and that the slight reversal of force which results has no disadvantages.

Curves without transitions.

Taking the leading bogie, we see that when it runs onto the circular curve, it is suddenly subjected to the acceleration $\frac{V^2}{R}$: to be accurate, this reaction is not instantaneous, because when the first pair of wheels of the bogie enters the curve, the increase in the acceleration is $\frac{V^2}{2R}$, and only reaches $\frac{V^2}{R}$ when the second pair of wheels runs onto it; but owing to the short distance between the bogie axles, the time which passes between the two pairs of wheels passing the tangent point is small. This variation in the acceleration is independent of the superelevation, the effect of which, in our opinion, is only felt when on the curve and not when running onto it.

We consider that the value of $\frac{V^2}{R}$ ought to lie between (6) and (7), which gives changes of acceleration of about 0.50. Messrs. **Baumann** and **Jaehn** recommend, when running over junctions, $\frac{V^2}{R} = 8$, and quote cases in which this value reaches 10 or more; this latter figure, however, appears to us to be exaggerated, not only as regards smooth riding but also from the point of view of safety.

We find that transitions must be used so long as regular speeds are worked to, as is proved moreover by experience.

To sum up, we consider that to appreciate the quality of a progressive transition curve from the point of view of smooth riding, the maximum change of acceleration of the movement of the bogie centres of the vehicle must be taken into account; the value of this

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(¹) According to our observations, a value of 0.25 can be used without difficulty.

change of acceleration can be calculated when $l = L$ (length of the transition curve = that of the superelevation ramp) in formula (10).

When the length of the transition has to be reduced, we can use the solution giving $L \geqslant 1 + 2a$ and calculate l from equation (13) in which the value of ψ may be made = 0.25 as maximum. The difference $L - l$ will be divided between the straight and the circular curve, with the reserve that the greater part of the excess is allowed to the curve, only a length equal to wheelbase of the vehicle being added to the straight.

In the absence of progressive transitions, the permissible speeds will correspond to the equation $\frac{V^2}{R} = 6$ to 7, and exceptionally we can have $\frac{V^2}{R} = 8$.

* * *

We should be very glad if railway engineers, who specialise in this matter, would express their opinion on these conclusions. We think the subject worthy of their consideration, seeing that the speed of railway vehicles is constantly tending to increase and that the unavoidable curves in the location of certain lines makes it necessary to have many transitions. A rational study of the subject would lead to much progress, to which railway engineers should contribute.

Madrid, April 1933.

[621. 33 (.44)]

Opening of the electrification of the Orleans to Tours line on the 19th July, 1933,

by JACQUES DUMAS, Engineer, E. S. E.

(*Le Génie Civil.*)

Mr. Appell, Assistant Secretary of State for Public Works, representing Mr. Paganon, the Minister for Public Works, attended the opening ceremony on the occasion of the electrification of the Orleans to Tours section of the main Paris to Bordeaux line of the Paris Orleans Railway Company on the 19th July 1933. Before describing the ceremony and giving details of the conditions under which this section has been electrified, we think it will be of interest to make it clear how far the electrification programme of this Company has been completed.

The Paris-Orleans Railway Company's electrification programme.

As everyone knows, the Paris-Orleans Company was the first of the great French railways to adopt electric traction; in 1900, it introduced electric locomotives for working trains in the Paris area between the old Austerlitz terminus and the new one at the Quay d'Orsay (5 km. = 3.2 miles). This form of traction had to be used when the Company moved the terminus for passenger trains to the Quai d'Orsay in order to avoid smoke in the underground tunnel connecting the two stations and running alongside the river Seine (1).

In 1904, the Company extended the electrification to its suburban trains running out to Juvisy (23 km. = 14.3 miles), but the investigation made in 1914 showed that the savings from electrification using current produced in steam power plants was insufficient to justify the investment of the large amount of capital required owing to the low cost of the coal burnt by the locomotives.

In 1917, at the instance of the Government who wished to reduce the importation of coal, the Paris-Orleans Company again investigated the question of electrification; it then prepared an electrification programme covering most of its lines in the « Massif Central » and asked for a concession to utilise the water power of the Upper Dordogne and its tributaries in order to supply power to the said lines.

When the matter was gone into more closely, the Company realised that the most interesting lines to electrify from the point of view of coal saving were those with heavy traffic or steep gradients. The idea of electrifying the secondary lines in the « Massif Central » was abandoned in favour, first of all, of the main line from Paris to Orleans, Vierzon, Limoges and Brive, then the cross country lines from Saint-Sulpice-Laurière to Gannat, and from Brive to Clermont-Ferrand with their varying profiles.

The first part of this programme,

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(1) See article on the extension of the line between the Austerlitz and Quai d'Orsay stations in the *Génie Civil* of the 25th January 1902.

which included the Paris to Vierzon section, was opened the 21st November 1926, a description being given in the *Génie Civil* of the 25th December 1926. As this first part of the electrification was completed before the first hydro-electric power station belonging to the Company (the Coindre station of 25 000 kw., situated on the two Rhues, tributaries of the Dordogne), the electric energy required for traction purposes was at first supplied by the Eguzon station (50 000 kw.), built by the « Union Hydro-Electrique » on the Creuse, and described in the *Génie Civil* of the 3rd July 1926.

In order to cover the supply of power required during the period of low water in the rivers in the « Massif Central », and at the same time make it possible to dispose of excess energy produced by the stations using these water courses in Winter, the Paris-Orleans Company entered into agreements with the principal producers of energy in the « Massif Central » and the distributors in the Paris region, to construct a high-capacity main transmission line between the « Massif Central » and Paris. This line, which was put into service on the 1st October 1932, and described in the *Génie Civil* of the 10th December 1932, starts from Marèges and ends at Chevilly, near Paris. The voltage is 220 000. The line collects the energy generated by most of the stations in the « Massif Central », in particular from the Eguzon and Coindre stations and there are two posts (Eguzon and Chaingy near Orleans) linking up with the 90 000-volt lines feeding the traction substations of the railway.

The operating results on the Paris to Orleans and Vierzon line having clearly demonstrated the benefits resulting from the electrification of heavily loaded lines, the natural course was to first electrify the lines with the heaviest traffic. The Company, therefore, decided to electrify the line between Orleans and Tours, the heaviest loaded after the Paris-Orleans section, before undertaking the electrification of the Vierzon-Brive section. The adoption of electric traction on the Orleans-Tours section will also make it possible to divert through Orleans part of the traffic of the line from Orleans to Tours via Vendôme, which is much more difficult as regards gradients.

As regards the electrification of the Vierzon to Brive section, it should be completed about the middle of 1935, as well as the two power stations at Marèges (130 000 kw.) on the Dordogne and the Cellette station (45 000 kw.) on the Chavanon, a tributary of the Dordogne, the concessions for which have been given to the Orleans Company at the same time as the Coindre station and which are intended to feed the Company's lines.

As regards the cross country lines from Saint-Sulpice-Laurière to Gannat and from Brive to Clermont-Ferrand, their electirification will probably be completed after that of the Tours to Bordeaux, Brive to Montauban and Tours to Nantes sections; these three main lines of the Company would in this way be wholly electrified and the two first will link up with the Midi system, largely electrified already as shown in the *Génie Civil* of the 27th August 1932.

Principal features of the electrification of the lines between Orleans and Tours.

The energy for the newly electrified line from Orleans to Tours is supplied by the Chaingy station, near Orleans, on the main 220 000-volt Marèges-Paris transmission line, by means of two 90 000-volt lines, feeding the five substations built at intervals along this section. Each of these lines consists of three aluminium conductors of 238 mm² (0.368 sq. inch) section and 20 mm. (13/16 inch) in diameter, supported by chains of six insulators on the straight, and of seven insulators at the suspension strain chains carried by steel pylons spaced 275 m. (900 feet) apart on the average. A galvanised steel cable, 60 mm² (0.031 sq inch) in section, secured to the top of the pylons, forms the earth wire; the minimum height of the overhead line above the ground is 9 m. (29 1/2 feet).

The 90 000-volt, 50-cycle current distributed in this way to the substations, is transformed there into continuous current by means of two groups, each consisting of two rotary converters, each of 2 000 kw. at 750 volts, arranged in series. The whole of the 90 000-volt side of these substations (busbars, circuit breakers, and transformers) is installed in the open, according to the arrangement already adopted for the substations on the Paris to Vierzon line; however, with the object of reducing the wages costs, the equipment of the new substations is entirely automatic.

Following the practice in the zone already electrified, the contact lines of the main line sections are overhead lines with compound catenary suspension, with double contact wires and alternated droppers. Each catenary line consists of : a catenary wire rope made of telephonic bronze 116 mm² (0.18 sq. inch) in section, an auxiliary catenary wire of electrolytic copper 104 mm² (0.161 sq. inch) in section, and two grooved trolley wires 107 mm² (0.166 sq. inch) in section. As an experiment, and in order to obtain information on the relative life, the contact wires on the down road are in hard electrolytic copper and those on the up road in cadmium bronze of high conductivity.

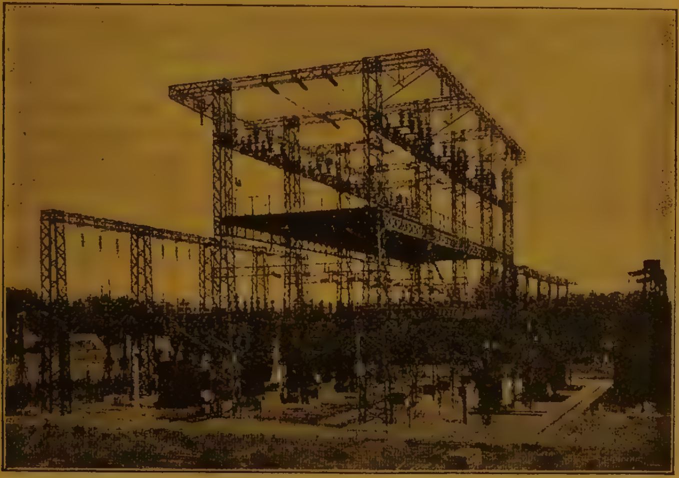


Fig. 1. — View of the open-air equipment of a substation.

As a result of the experience obtained on the Paris-Vierzon section, everything possible has been done to make the two tracks independent of one another. With this object, the catenaries of the two tracks, instead of being suspended from a portal, are carried on two entirely independent pylons. These pylons have been built up of U bars assembled by cross braces, and this has resulted in an appreciable saving of metal as compared with the portal supports of the old equipments. The normal distance apart of these supports on the straight is 63 m. (207 feet).

In the stations, where it would be impracticable to erect independent supports for each track, the supports have been kept away, as far as possible, from the points and crossings, on which derailments are to be feared during shunting operations, by using either brackets for two tracks or portals with stayed bracket supports, or semi-rigid portals formed by two layers of cables capable of carrying, without undue deformation, widely differing loads.

Unlike what had been done on the Paris-Vierzon section, the device adopted to limit the movement of the contact lines under the action of wind, instead of being by transverse cables, is by means of tubes, adjustable in length and secured by insulators at one end to the auxiliary carrying cable of the catenary suspension, and at the other end to the pylon or bracket serving one line only. On curves, stretchers keep the trolley wires on the same vertical plane as the auxiliary carrying cables.

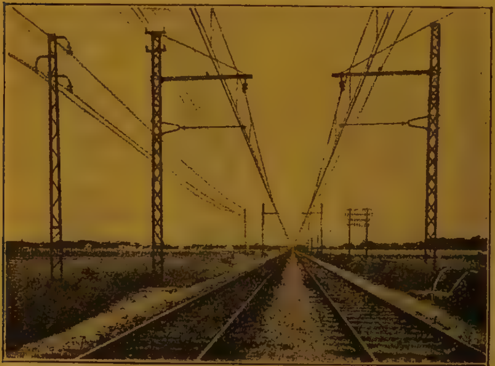


Fig. 2. — Equipment of the running lines on the straight.

The lines are divided into sections, the ends of which are arranged to fall at each substation and half-way between two consecutive substations. The continuity at the section points is obtained by means of ultra-rapid circuit breakers provided either in the substations or at the section posts along the track. The circuit breakers in the substations and the section posts belonging to a given section of line are closed or opened together by means of a special 1 500-volt circuit (pilot wire).

The daylight signalling and the automatic block equipments are fed by a 10 000-volt line placed within the boundaries of the railway, which is fed at both ends in the outer substations through a 90 000/10 000-volt transforming station and a standby equipment consisting of a motor generator with a battery of accumulators. A booster transformer placed in an intermediate substation is provided so as to maintain over the whole length of the 10 000-volt line a suitable voltage when the line is only fed at one of its ends.

In order to cover the electrified working on the Paris-Vierzon section, the Paris-Orleans Company had 80 rail motor coaches in the suburban service, 200 locomotives for the passenger and goods services, and 5 high-speed test locomotives which were described in the *Génie Civil* of the 12th March 1927. As a result of more than six years operating experience, the test locomotives which gave the best results were found to be the two of the 2D2 type, numbered E 501 and 502, the electrical equipment of which was supplied by Messrs. Brown-Boveri and Co., and the mechanical part by the Winterthur Locomotive Works; these locomotives were described in detail in the *Génie Civil* of the 4th June 1932.

These locomotives have two driving axles with wheels 1.500 m. (4 ft. 11 in.) in diameter, a leading bogie, and a control cab at each end; their horsepower is 3 600 on the hourly rate, and 3 200 under continuous load; their total weight is 125 t. (123 Engl. tons), and their adhesive weight 72 t. (70.8 Engl. tons).

In order to bring up its stock of locomotives to the required figure, in view of the opening of the Orleans to Tours line, the Paris-Orleans Company ordered from the « Compagnie Electromécanique » and « Compagnie de Fives-Lille » group 25 locomotives of this type, which will be numbered E 503 to 527, for delivery in 1932. These locomotives only differ from the preceding ones in such alterations in details as experience had shown to be needed : the diameter of the driving wheels has been increased to 1.75 m. (5 ft. 9 in.), the power at the hourly rating and continuous rating have been raised to 3 700 and 3 300, and the total and adhesive weights to 140 and 80 tons.

In addition, in view of the opening of the Vierzon to Brive line in 1935, the Paris-Orleans Company have ordered four new high-speed trial locomotives, of the 2D2 type, which will be numbered E 701 to 704, the power of which will be 4 100 at the hourly rating and 3 600 at the continuous rating.

In order to appreciate the progress made, it should be noted that the electrification of the Orleans to Tours line respectively raises from 238 to 353 km. (147.9 to 219.3 miles), and from 970 to 1 265 km. (602.7 to 786 miles), the respective length of the lines, and of the single track of the electrified system of the Paris-Orleans Company, which will increase from 5 to 6.4 million tkm. gross (3.15 to 4.04 million Engl. ton-miles) the yearly traffic worked electrically. The consumption of energy by the railway will increase from 115 to 143 million kw.-hours, and the coal saved, from 260 000 to 318 000 tons. This electrification has made it necessary to increase the number of substations from 11 to 16, their total power from 55 000 to 75 000 kw., and the length of the 90 000-volt lines feeding them, from 640 to 865 km. (397.7 to 537.5 miles).

Opening of the new line to electric traction.

On the occasion of the opening of the Orleans to Tours line to electric traction, the Paris-Orleans Company ran a special train on the 19th July, consisting of Pullman coaches belonging to the International Sleeping Car Company, and hauled by the high-speed electric locomotive No. E 501. This train, which was booked to cover the run from Paris (Austerlitz) to Saint-Pierre-des-Corps (Tours) in two hours, i. e. at a speed of 115.5 km. (72 miles) an hour, started from the Austerlitz station at 2 p. m. and stopped in St-Pierre-des-Corps Station at 3.56 p. m., i. e. four minutes before the booked timing, after having covered the distance at the average speed of 120 km. (75 miles) an hour, in spite of the speed reductions required when running through the stations at Juvisy, Etampes, Les Aubrais-Orleans, and Blois, having reached, at places, speeds of nearly 150 km. (93.2 miles) an hour, a record much to the credit of the Paris-Orleans Company. It is well to add that, during the official trials which took place a few weeks beforehand, this Paris-Tours run was made in 1 h. 50 m., i. e. at an average speed of 126 km. (78.3 miles) an hour, a speed higher than that of the « Cheltenham Flyer » in England, which, during the trial runs in September 1931, covered the distance of only 124 km. (77 miles) between Swindon and London at the speed of 125 km. (77.7 miles) an hour.

At Saint-Pierre-des-Corps, the guests of the Paris-Orleans Company were taken by motor car to visit the substation at Epines-Fortes, the last on the line, and were welcomed there by Mr. Richemond, President of the Board of Directors, and the higher officials of the Paris-Orleans Company. After having broadly sketched the electrification work, Mr. Richemond reminded his listeners that, whilst carrying out its schemes for electrification, the Paris-Orleans Company, far from neglecting steam traction, had just transformed some *Pacific* locomotives purchased before the War by modernising them in such a way that their power had doubled at the same time that their steam consumption per H. P. had been reduced by more than 30 %. These old locomotives had in this way been more efficient and powerful than recent engines.

Replying to Mr. Richemond, Mr. Appel, Under Secretary of State for Public Works, pointed out that the electrifica-

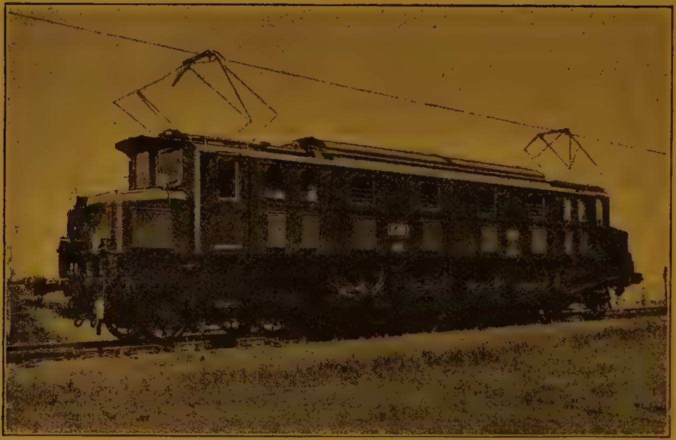


Fig. 3. — View of one of the new high speed locomotives,

tion, first undertaken by the railway companies to reduce the cost of coal imported from foreign countries, was now directed towards equipping electrically the whole of the country, by making electricity available in the country districts so as to bring them more comfort and greater well-being.

Mr. de Boysson, Assistant Manager of the Paris-Orleans Company, then gave a technical review of the electrification of the new line, in connection with which an endeavour had been made to facilitate the maintenance work by reducing to a minimum the parts under voltage, and by perfecting to the furthest possible extent the thousand and one details of the equipment of the track, the importance of which will be appreciated when it is noted that the overhead line had to transmit to locomotives running at speeds of 120 km. (75 miles) an hour current of 2 000 amperes at 1 500 volts. Nonetheless, the electrification of this line had only taken 18 months, that is three months less than the time expected, thanks to the way the contractors had carried out their work : the « Als-Thom » Company for the locomotives and electrical equipment; Messrs. « Drouard Frères » for the contact lines; and the « Enterprise Industrielle » Company for the 90 000-volt lines.

After visiting the Epine-Fortes substation, the Company's guests returned by motor to Tours, from whence they returned to Paris by train.

621.43 (.42)

130-H.P. oil engine-driven railcar for the Great Western Railway (Gt. Bn.).

(*The Engineer.*)

At the end of July 1933, we were able to give a short illustrated description of the chassis of an oil engine-driven railcar manufactured by Hardy Motors, Ltd., of Southall, and fitted with a 130-H.P. A. E. C. oil engine. At that date the vehicle was still more or less in the experimental stage, and some of the smaller details of its design were not definitely settled. Nor were the makers ready to publish details of the body which was to be fitted to it. The experimental running of the chassis has now been satisfactorily completed, and as may be inferred from the engraving, figure 1, the body has been fitted. The complete vehicle, which has been shown at the Commercial Motor Transport Exhibition at Olympia, was purchased by the Great Western Railway Company, to enter service, probably between Slough and Reading.

The coach, as will be observed from figures 1 and 2, is of a stream-lined form. It was made and fitted to the chassis by Park Royal Coachworks, Ltd., of Abbey-road, Park Royal, N. W. 10. The reduction of windage resistance and judicious saving of weight in the design of the chassis has made it possible for the 130-H.P. engine to give the vehicle a satisfactory performance. Carried on two four-wheeled bogies, the car is 62 feet long by 9 feet wide overall, and weighs less than 20 tons. The power-weight ratio is therefore higher than it is on most railway trains or vehicles. Besides a luggage compartment at one end, the coach has accommodation for sixty-nine passengers, but we understand that as many as eighty-eight could be carried if the luggage compartment was not present and the seating rearranged. The important influence of the smooth-

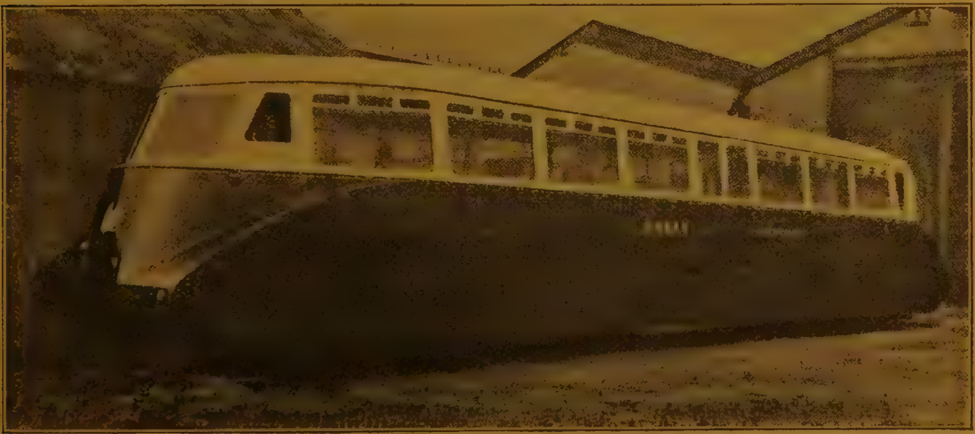


Fig. 1.

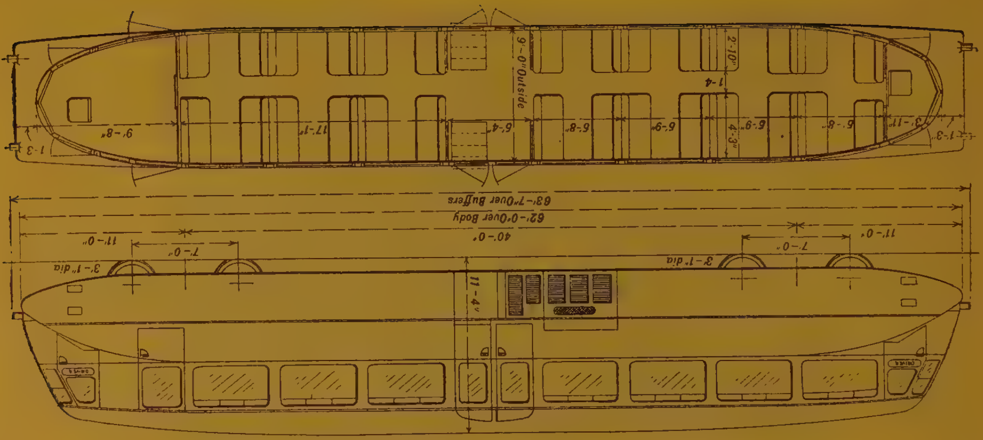


Fig. 2. — General arrangement of railcar.

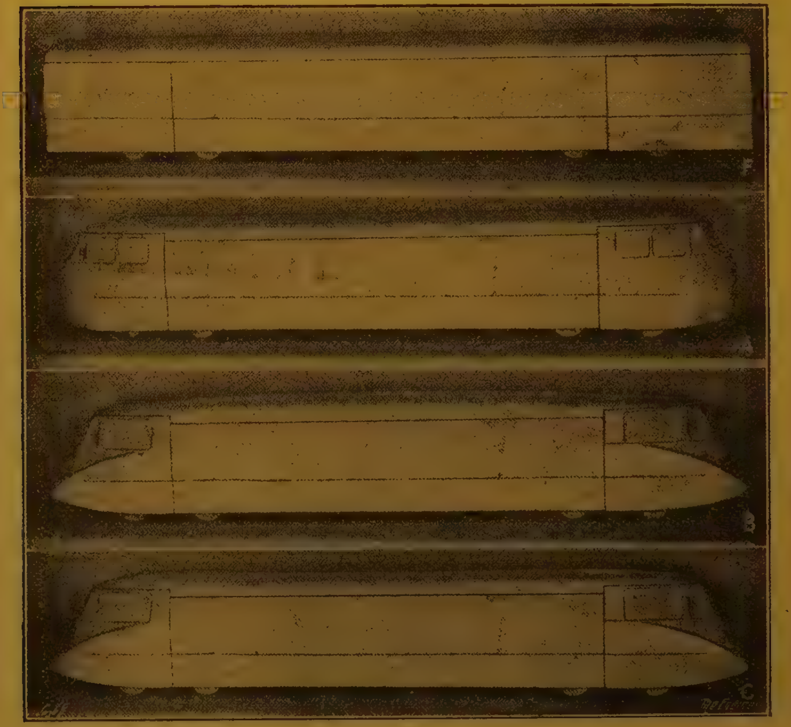


Fig. 3. — Models for use in wind tunnel tests.

ness of the exterior of the body on the windage resistance has been fully realised, and there are practically no external projections whatever. Even the steps to the driver's cab from rail level have been arranged to be covered by flaps when the door above them is closed, and hinges which must project of stream-line form. The coach is painted in G. W. R. colours.

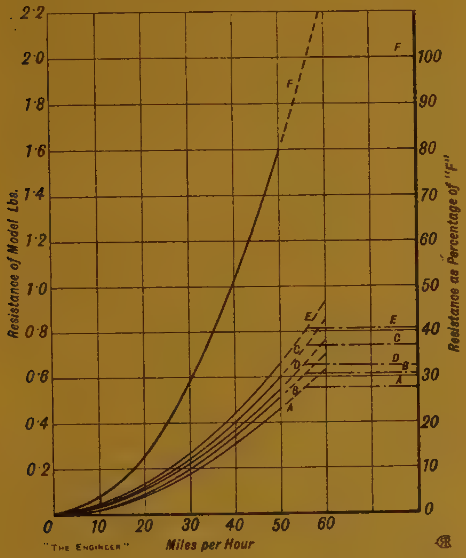


Fig. 4. — Results of tests of models in wind tunnel.

It was designed to follow out the results of wind tunnel tests on a number of models, some of which are illustrated in figure 3. A graph (fig. 4) the curves upon which are lettered to correspond with the model photographs in figure 3, shows the wind resistance in pounds of the models plotted against speed of motion. The curves have been extended beyond the results of the actual wind tunnel tests themselves, as shown by the dotted lines, in order to judge of the performance of the vehicles at higher speeds. Curve F, which shows a very high resistance, applies to a coach with flat ends of conventional form, except for the shrouding of the wheels. It was taken in order that some basis for the comparison of the stream-lined vehicles might exist. The best results were achieved by the model A, but, as we think our readers will agree, a vehicle constructed to have the shape of this model would be frankly ugly. A design which would give comparable results and at the same time have a pleasing form was therefore sought, and models B and C were the result. At first glance, these two might be taken to be identical, and, in fact, the only difference occurs in the shape of the roof, which in B is curved upwards at the centre 6 inches relatively to C. The surprisingly large influence of this small change of shape may be traced upon the graph. As might be inferred from figure 1 at the head of this article, it was decided to construct the body in accordance with the shape of model B. The other curves on the graph refer to models of other forms examined by the makers, but since they had no influence on the final design adopted, they need no further reference here.

The chassis.

The design of the chassis of this railcar is unusual in that the engine is placed within the body and to one side of the frame near the middle of its length. Regarding, for purposes of description, the driving bogie end as the rear end of the double-ended coach, the engine is situated on the left-hand side of the frame a little forward of the central position. The arrangement will be made clear by a reference to the photograph and drawings reproduced in figures 5 and 6. Both axles of the rear bogie are driven and the whole transmission system lies outside the main frame on the left-hand side of the vehicle. The mechanism is, however, hidden from view in the completed coach by the shrouding which extends to within a few inches of the rails. As far as possible, consistent with the best modern practice in railway carriage construction, weight has been saved in the main frame and bogies, so that the total weight of the chassis is only 13 tons 10 cwt. The general design may be followed from the drawings, figure 6. Before a choice was made consideration was given to the advantages of several different types of bogies and a design was finally decided upon that could readily be adapted to suit both the braking and driving systems it was intended to employ. As will be seen from the drawings, each bogie is of the swing link bolster type, with laminated springs over the axle-boxes and an auxiliary rubber suspension block, and nests of coil springs between the bolster and spring plank. The main frame allows the engine and the various units of the transmission system to be conveniently carried by outrigger members cantilevered from it. An additional advantage of the construction is that convenient steps from rail level into the body of the car at the central vestibule can be built in without the necessity for any projection that might increase the air resistance of the vehicle. A « trap door » in the floor may be raised and steps have been built in below the floor and clear of the frame to rail level. A panel on the body opposite the steps is hinged and so connected with the « trap door » that it opens outwards as the flap is raised. The main members of the frame are 10-in. by 3 1/2-in. channels trussed with 4-in. by 4-in. by 1/2-in. angles.

Engine and transmission.

As a result of careful stream-lining of the body and judicious saving of weight in the chassis, the 130-H.P. A. E. C. oil engine, similar to that fitted in many commercial road vehicles, provides ample power. The radiator for the engine is mounted conventionally in front of it. Since, however, the whole engine unit is enclosed within the bodywork of the coach, special measures had to be taken to cause a flow of air through the radiator. Beneath the car there is a swinging sheet metal flap hung on a vertical axis. Its free end projects into the mouth of a sheet metal duct, and it is disposed more or less across the frame of the vehicle. Naturally, when the vehicle is travelling this flap swings backwards until its further motion is prevented by rubber buffers on the mouth of the duct, and in this position it deflects air into the duct, irrespective of the direction of travel of the coach. The air is thus guided into the space in front of the radiator, through which it is drawn by a fan in the usual manner, and after passing over the engine, it is discharged through louvres in the side walls of the coach. Owing to the low shrouding of the vehicle, the arrangement has the advantage that the heated air discharged

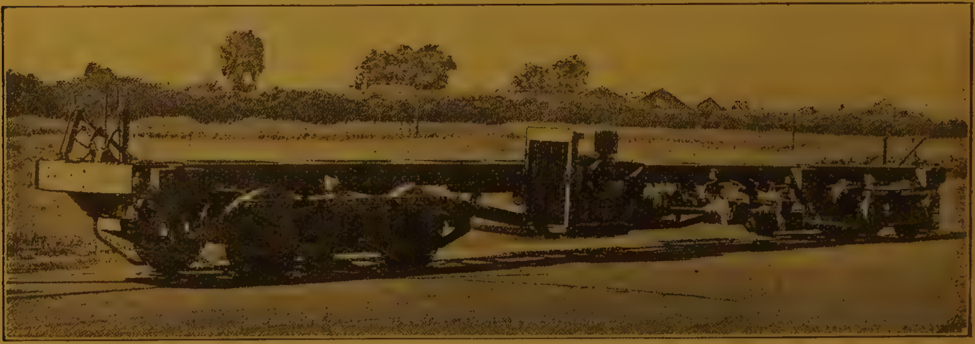
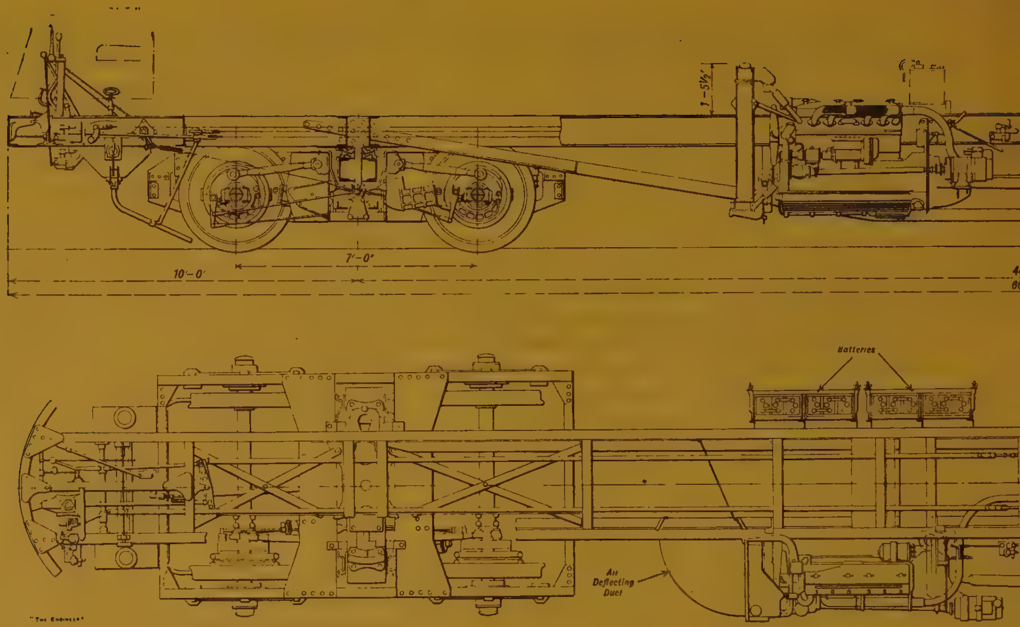


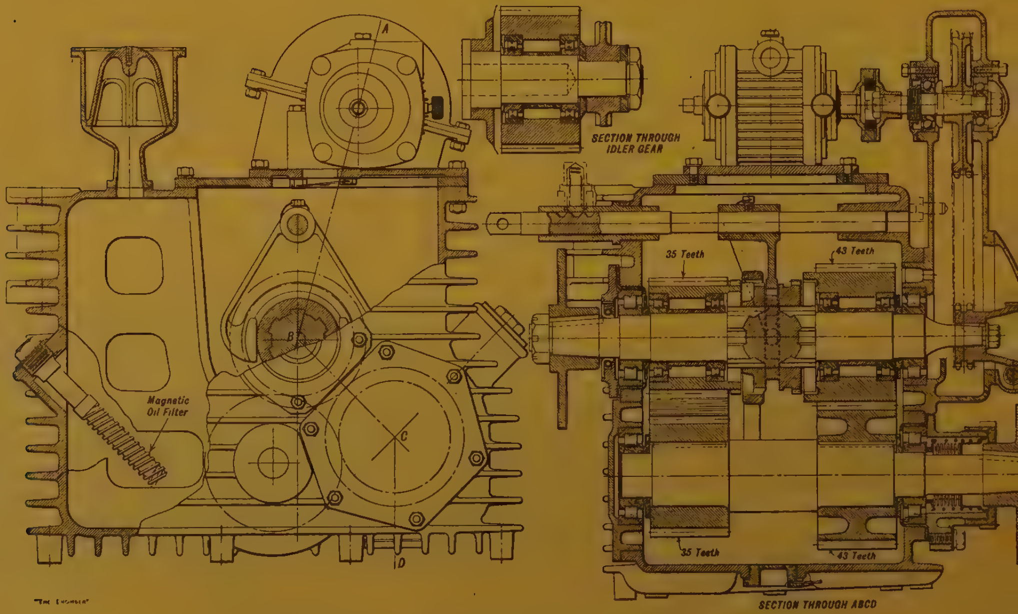
Fig. 5. — Side view of chassis showing arrangement of power unit and transmission.

cannot again find its way to the inlet to be recirculated. It is obvious therefore, that in whichever direction the railcar is travelling, a plenteous supply of cool air will be drawn through the radiator.

Immediately behind the engine there is a fluid flywheel, which takes the place of a clutch. Thence a shaft with two universal joints takes the drive to the pre-selective gear-box of the Wilson type. It is at this point that descent from the interior of the coach to rail level is made, and the shaft is so placed that it does not interfere with the construction of the steps mentioned above. The gear-box gives four forward speeds and a reverse, the latter gear being retained owing to its convenience when short shunts are required. Speeds of 14.45, 25.4, 40.25, and 60 miles per hour are obtainable by the use of the forward gear ratios when the engine is turning at 2 260 r. p. m. Beyond the Wilson gear-box the next mechanism to deserve notice is the reverse gear box, a section through which is reproduced in figure 6. In whatever direction the vehicle is travelling there is no conventional direct drive, as a study of the engraving will demonstrate at once. The output shaft is situated below and further from the main frame of the vehicle than the input shaft and connected with it by constant mesh gears. A dog clutch engages the input shaft, upon which it rides, through two 43-toothed intermeshing gears, or alternatively through a 35-toothed wheel, an idler gear, and another 35-toothed gear, with the output shaft. There were two reasons why a direct « straight-through » drive was not adopted for travel in one direction. Firstly, of course, as will be clear to all those who examine the general arrangement plan of the chassis, it was necessary to situate the output shaft further away from the frame than the input shaft in order to take the drive to the axles in the manner shown. But of far greater influence was the fact that, given a « straight-through » drive in one direction, no less a number of gear wheels than five is necessary to obtain reverse. An arrangement by which two gear wheels were engaged for the drive in one direction and three for the drive in the other involved no greater number of gear wheels and made no such great distinction between the efficiencies of the two drives. The aluminium casing of this gear-box is ribbed to bring about efficient cooling and provides a large oil capacity at a low oil level — a point of some consequence, as a high oil level leads to churning and to rapid heating of the oil. At the rear end a chain drive is taken from the output shaft to a rolling drum exhauster mounted on the top of the gear-box. This exhauster maintains the vacuum in the reservoirs of the brake system and is additional to another exhauster driven directly by the engine. It is provided to maintain the vacuum when, for instance, the vehicle is coasting down a long hill with the gears of the « Wilson » box in neutral and the engine idling, under which conditions the engine-driven exhauster is more or less out of action owing to its low speed. The exhausted air is delivered into the gear-box, whence it escapes through a specially designed breather adapted to trap any oil that may be entrained in the air and return it to the sump. The arrangement is such that the gears in the box act as « oil throwers » to lubricate the moving parts of the exhauster.

Up to this point it has only been necessary to provide universal joints in the connecting shafts between the units of the transmission system to allow for differences of angularity. In the connection between the reverse gear-box and the driving axles, however, a further degree of flexibility is necessary in order to allow for rotation of the bogie round its centre pivot. The design of the telescopic shaft forming this link in the system may be seen in the drawing, figure 7. It is carried between two universal joints to provide for the necessary angular movement and consists as to its central

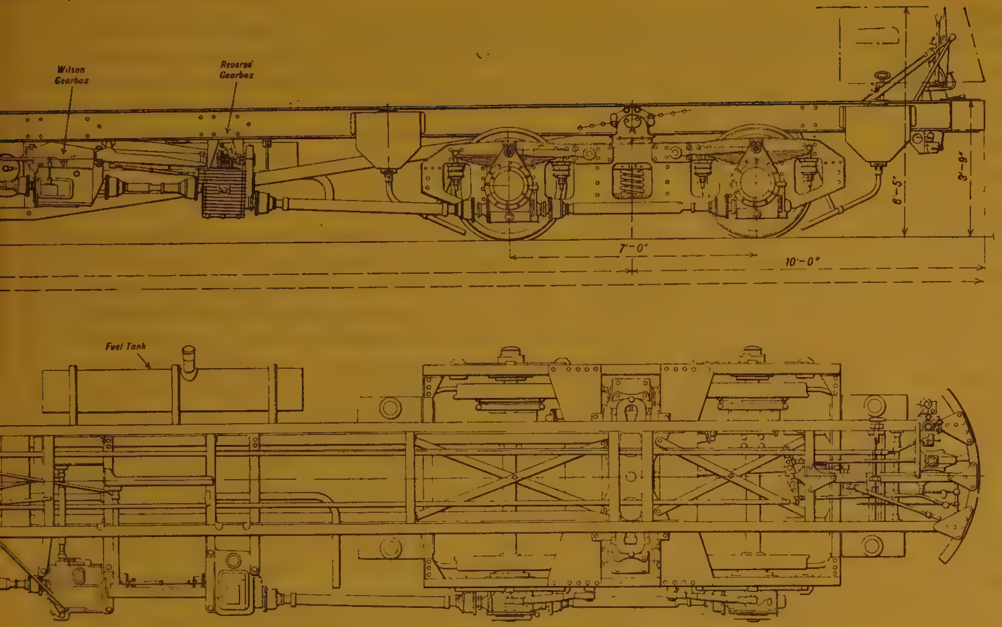


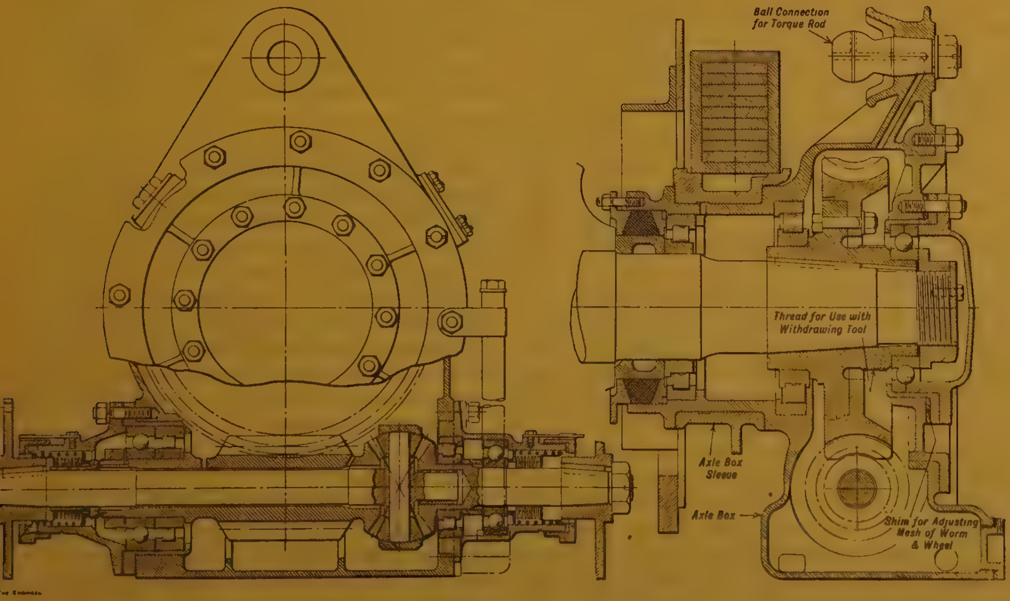


Reverse gear box and exhauster for vacuum brake.

Fig. 6. — Chassis of 130-H.P. oil engine-driven rail

ment of complete chassis.




Ball Connection for Torque Rod
Thread for Use with Withdrawing Tool
Axle Box Sleeve
Axle Box
Shim for Adjusting Mesh of Worm & Wheel

Worm drive and axle box assembly.

service on Great Western Railway.

part of a tube 3 1/2 inches in external diameter. Into this tube at each end there is fitted the « butt » of a solid rod having a shoulder of the same diameter as the tube. The two are welded together at the shoulder. Another shorter tube of the same diameter fits over the rod on the other side of the shoulder and is similarly welded into position. The rod is of such length that it projects beyond the outer end of the second tube and over the greater part of its length is of such a diameter as to provide a wide clearance within the bore of the tube. Upon splines cut on the projecting part of the rod a hollow shaft is adapted to slide, and, being of smaller diameter than the internal diameter of the tube, its end can project into the clearance space between the rod and the tube. The outer ends of these two hollow shafts connect with the universal joints at each end of the telescopic shaft assembly. As will be seen from the drawing, covers screwed on to the ends of the tubes and nuts screwed on to the ends of the shafts prevent the complete withdrawal of the shafts from within the tubes, but dismantling of the assembly is easily performed by the removal of the caps, which are held in position by locknuts and contain a sealing ring to prevent leakage of the lubricant. A motion of 4 1/4 inches is provided at each end of the assembly and lubrication is obtained by filling the clearance spaces with grease.

The next mechanism to be described is particularly interesting, and the true merits of the design cannot be appreciated without considerable study of the drawing reproduced in figure 6. Both axles of the rear bogie are driven — see also figure 8 — through worm and wheel mechanisms, and as any difference in the diameters of the wheels would set up undesirable stresses in the shaft between the two worms an interaxle differential gear is housed in the casing of the leading axle-box. The drawing reproduced in figure 6 refers to this leading axle. In general, the design of the worm drive to the other axle is the same except for such differences as the omission of the differential brings about. The main feature of the design, which is covered by patents, is the manner in which the worm is carried in the axle-box casing.

From the flange which is attached to the universal joint of the telescopic shaft the drive is taken to the central member of the differential gear. One of the outer members of the differential passes on the drive to the second axle, while the other is solid with the worm driving the first axle. The neat manner in which the various parts are firmly supported is better discovered by examination of the drawing than it can be described here. The worm wheel is bolted and splined to a flange carried on the axle itself, and the manner in which its mesh with the worm is adjusted deserves mention. It will be observed that there are two end covers for the worm wheel assembly, the smaller bolting into the larger. A ball thrust and journal bearing is carried by the smaller of these two covers. Since the large roller bearings that transfer the load of the vehicle on to the axle are so designed as to have no locating properties, it is this ball bearing that locates the casing relatively to the axle. Thus by placing shims between the smaller cover and the face to which it bolts the position of the axle relative to the axle-box — or, in other words, the position of the worm wheel relative to the worm — may be altered. Another important feature of the whole design becomes apparent after a little study of the drawing. The whole of the worm and wheel mechanism may be dismantled for repair, replacement, or examination without even relieving the axle of the weight of the vehicle. Removal of the larger cover allows the worm wheel to be withdrawn, while the worm assembly is easily removed when the end covers of its casing have been taken away. Moreover, all these covers are situated accessibly at the side of and not beneath the vehicle.

As we have already noted, the axle rotates in roller bearings within the axle-box. An arm projecting from the top of this box is attached by a ball-ended lever to the end of a torque rod. The springs, therefore, which are clamped to the axle-box sleeve take none of the torque. A small detail indicative of the care and thought which has been put into the design of the railcar is illustrated by the accompanying sketch, figure 9. Here the large circle indicates the wheel. The axle-box and its guides, together with the torque rod, are also shown diagrammatically. Supposing the vehicle to be travelling towards the right, the torque reaction will be anti-clockwise and the torque rod, attached above the axle, in tension. Thus the force in the torque rod tends to pull the axle-box against just that set of guides on which the pressure to drive the vehicle along is being exerted. Where the torque rod to be connected below the axle, as indicated by the dotted lines, its effect would be in opposition to the driving pressure, and the arrangement might well give rise to chatter at the driving face of the guides. The axle-box itself is also mounted in a sleeve so that the thrust on the guides may be truly normal. Otherwise one of the effects of torque reaction would be to give the box a tendency to cant in the guides and bear only on the corners.

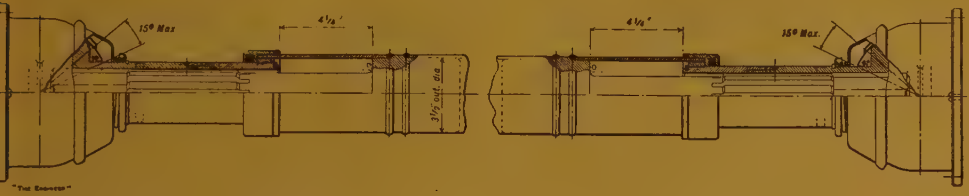


Fig. 7. — The telescopic shaft between reverse gear-box and final drive.

Brake gear.

In the brake gear a departure has been made from conventional railway practice by the fitting of internal expanding brakes similar to those of a road vehicle. Between road and rail vehicles, however, there is the essential difference as far as brake gear is concerned that in the one case the wheel may be readily removed, while in the other it may not. The mechanism is so arranged on this railcar that access can be gained to the shoes, operating cams, and pivot pins without removing the drums or wheels. The whole assembly, a drawing of which is reproduced in figure 10, is mounted on the axle behind the wheel. The drums themselves, split to be removable for truing up or replacement, are bolted and registered into the wheels, while the rest of the gear is mounted on a carrier riding on roller bearings on the axle and prevented from rotating by a torque rod. A split collar clamped to the axle retains the carrier in the correct position and holds it up against the face of the wheel. It will be clear from this description that it is only necessary to slack back the retaining collar, detach one end of the torque rod and disconnect the brake rod in order that the whole carrier may be slid back along the axle and the various parts mounted upon it inspected.

Two vacuum cylinders are mounted on each bogie. Each is independently connected to the brake cam shaft lever of one axle, so that complete compensation of the brakes is ensured. A cross-shaft of compensating type coupled to the hand brake lever in the driver's compartment is also connected to the brake cam levers. Vacuum for the brakes — and also incidentally for the sanding gear — is maintained by two exhausters,

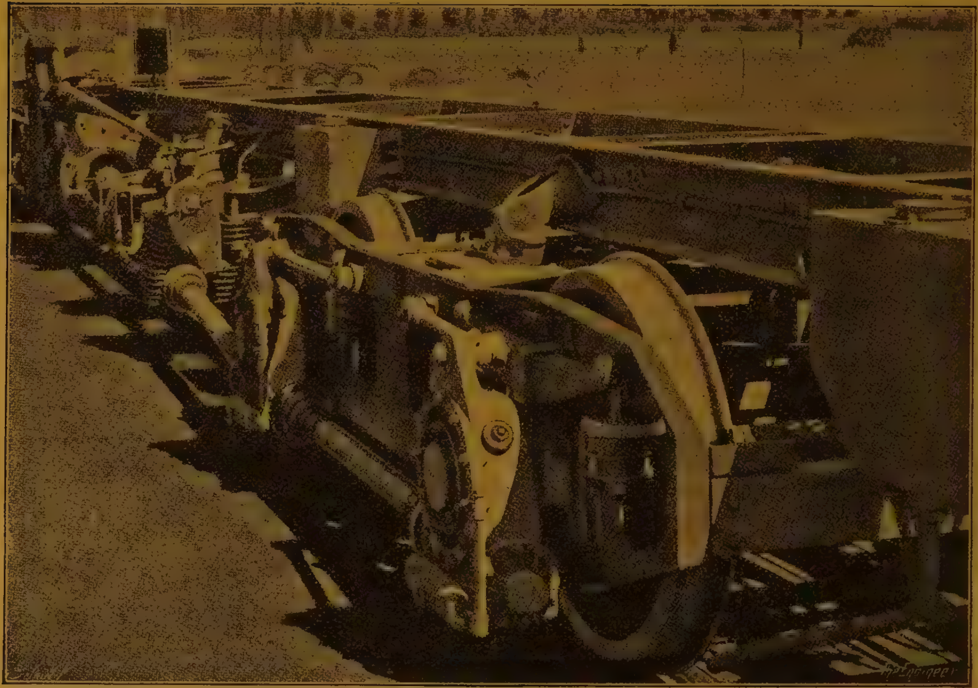


Fig. 8 .— The driving bogie.

both of which have already been mentioned. Three reservoirs are mounted beneath the chassis, and gauges in the driver's compartments show the vacua in these reservoirs and in the brake pipe.

Before passing on to describe the controls, it should be noted that a large-capacity electric accumulator battery is mounted on the frame just opposite to the engine, while the fuel tank is on the same side as the accumulator, but further back.

Controls.

The controls under the charge of the driver are arranged in an identical manner at each end of the vehicle. They comprise an accelerator pedal, a preselective gear lever, a reverse gear lever, a ratchet hand brake, a finger-control lever for the vacuum brake, and another for the sanding gear. Lastly, there is a pedal for changing gear in the Wilson box, for it will be recalled that in this type of box the next gear to be engaged is pre-selected, but not actually engaged until a second control is operated. The hand brake lever at each end of the coach only applies the brake on the wheels of the bogie immediately beneath it. The two levers are thus not interconnected. In the case of the reverse gear lever and the preselective gear lever, however, interconnection is necessary. It is obtained by means of steel ribbons attached to the ends of double-ended levers mounted beneath the coach and rotated by the controls. The arrangement is such that whatever the direction of rotation one of the ribbons is in tension. Approximatley half-way along the vehicle a double-ended lever is mounted between the two ribbons and to it is attached a rod making connection with the mechanism controlled. A similar arrangement has been adopted for interconnecting the gear change pedals at each end of the car with the Wilson gear-box. As, however, the « working » movement is always in the same direction and a spring can be arranged to give the return motion, only one steel ribbon is required, and it is always in tension.

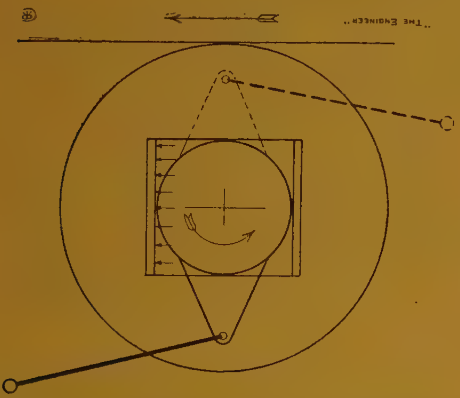


Fig. 9. — Diagram of torque rod connections.

As the gear-selecting lever works in a « gate », trouble might arise through the gate at one end of the vehicle interfering with the free motion of the lever at the other. To make any such state of affairs impossible, only one gear-selector lever is supplied. It is so arranged — see figure 11 — that it may be lifted out of its socket and, when the direction of motion of the car is reversed, carried from one end of the coach to the other. It is the toe of this detachable lever that engages with the gate, so that when it is removed no interference is possible. There are only four slots in the gate corresponding to the four forward ratios available. Reverse is obtained by lifting the lever slightly to clear the gate and drawing it back as far as it will go.

The coach is very simple to drive. An electric starter brings the engine into action. The reverse lever is next thrust in the direction in which it is desired to travel, and the pre-selective gear lever is brought through the gate to the « first » or « second » gear position. Then, on pressing down the gear change pedal and releasing the air brake or hand brake, the railcar moves smoothly away. To engage another gear — whether higher or lower — the pre-selective lever is moved once more, and the gear change pedal is again depressed at the moment when it is desired the actual change shall be made.

This railcar, although its body is of streamlined form, has not been designed to travel at very high speeds, but to give economical services at speeds up to 60 m. p. h. It is, in fact, intended for suburban rather than express long-distance service. In order that its maintenance costs may be small, as many parts standard to Hardy road vehicles have been used as the design will permit, while the various units have been placed in accessible positions so that the replacement

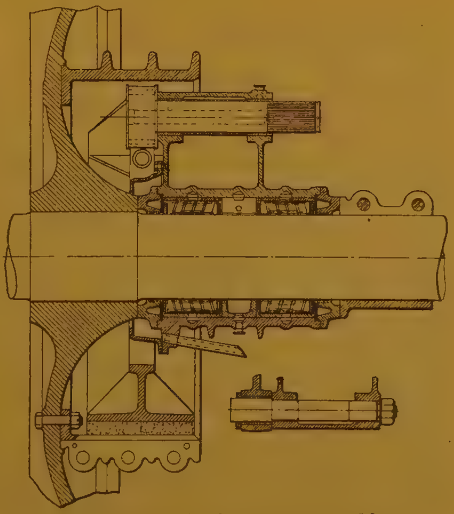


Fig. 10. — Brake gear assembly.

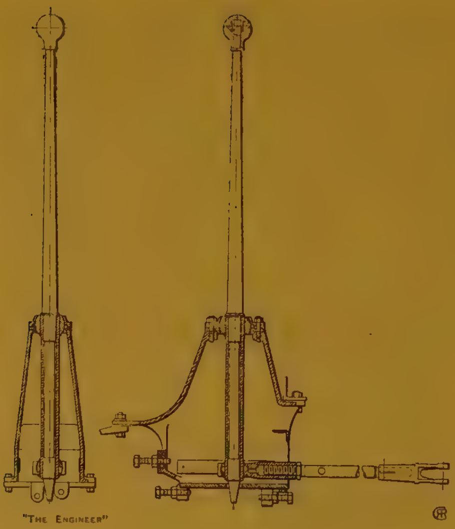


Fig. 11. — Arrangement of gear selector lever.

of worn parts may be carried out easily and quickly. As to the interior of the body, it is particularly luxuriously furnished in a colour scheme of brown and green; the seats are upholstered in leather, and the passengers sit back to back. At one end a door leads into the luggage compartment, but at the other glass panels are interposed between the driver and the interior of the coach. As long windows are arranged on each side, the coach has a visibility as good as that of an « inspection » car. Deep luggage racks are provided and lights mounted beneath them are arranged above the heads of seated passengers in just that position most suitable for reading. A recording speedometer is open for inspection by the passengers; there is a clock, and in case of emergency a handle may be depressed to apply the brakes and stop the flow of fuel to the engine.

[621. 43]

Transmissions for Diesel locomotives and rail-cars,

by STUART MIALL, B. Sc.

(From *Diesel Railway Traction*, Supplement to *The Railway Gazette*).

The Lysholm-Smith transmission.

During the past 15 years many attempts have been made to obtain by hydraulic means a widely varying output torque from the more or less fixed input torque of the internal-combustion engine. In some instances a variable-capacity pump has been used te deliver oil to a plunger type of hydraulic motor, the fluid pressure and the motor torque rising as the quantity delivered and the motor speed were diminished; in other instances rotary pumps and turbines have found favour. That the torque of a prime mover could be multiplied by a combination or hydraulic turbine elements was demonstrated long ago by Föttinger's hydraulic speed transformer, a contrivance used in vessels propelled by steam turbines in the days when mechanical gearing was too imperfect to be trustworthy, but this transformer was not designed to provide more than one ratio between primary and secondary torques. For automobile and railway work several carefully chosen ratios, or a ratio infinitely variable between specified limits, must be provided.

In Föttinger's torque converter there was necessarily a stationary reaction member. Later developments have proceeded on divergent lines and may be classified as under :

a) Those in which the reaction member has been removed, a one to one torque ratio secured, and change-speed mechanical gearing incorporated;

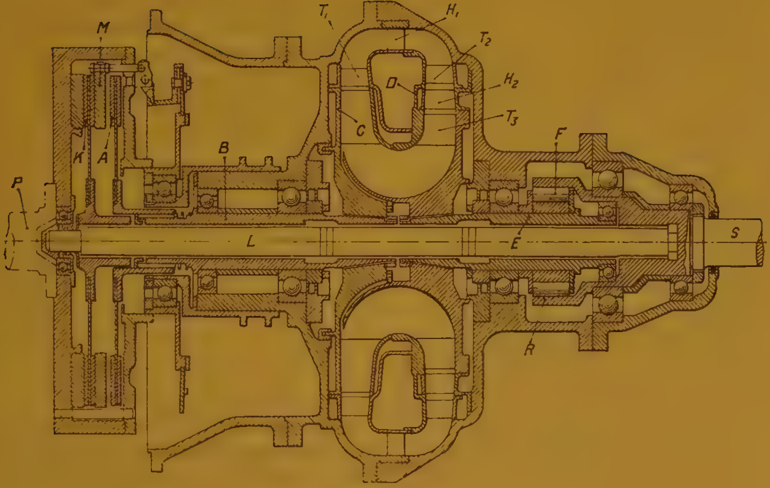


Fig. 1.

b) Those in which the stationary reaction member has been retained and the blading arrangements modified to give an increasing secondary torque when the secondary speed is reduced, the primary torque and speed meanwhile remaining more or less constant.

Under heading *a*) come the Daimler transmission (the fluid flywheel with the pre-selective epicyclic gearbox) for automobiles, and the various combinations with gears of the Vulcan-Sinclair hydraulic coupling, which have been described from time to time in the *Railway Gazette* (1). Under heading *b*) come the Vickers-Coats torque converter (described in *The Railway Gazette* for March 25, 1932) (1) and the Lysholm-Smith torque converter. The latter was described by S. André and G. Göransson in a paper before the recent World Power Conference, and the accompanying illustrations are taken from their paper.

Method of operation.

Figure 1 shows the sectional arrangement of a Lysholm-Smith torque converter suitable for a railway application. It will be observed that no provisions are made for reversing the direction of rotation of the secondary shaft, but this is not necessary because the right-angle drive serving to rotate the axle of the car has the usual sliding-dog arrangement for selecting one of two loose gears.

The engine fly-wheel contains a double-purpose clutch by which the engine shaft P may be connected either to the pump member C of the torque converter via sleeve B, or via intermediate shaft L to the output or secondary shaft S. The centre plate M is pressed by the action of the springs to the right or to the left, thus picking up fabric-faced discs A or K. At starting, disc A is gripped and pump impeller C sets up a rapid circulation of the fluid within the torque converter. Leaving C, the fluid moving outwards (and circumferentially) impinges on the first ring T_1 of the secondary moving blades. It is then directed by a bladeless cavity through the first ring of stationary blades H_1 and by a second bladeless cavity through T_2, H_2 and T_3. Blade rings T_2 and T_3 are attached to the secondary member and revolve with T_1, while H_2, like H_1, is a fixed ring of blades. On leaving T_3 the fluid is drawn again into the pump impeller C.

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(1) The Vulcan-Sinclair hydraulic coupling and the Vickers-Coats torque converter are described hereafter.

When the secondary member first starts to revolve its torque output is about five times that on the primary or pump member. This is equivalent to the low-gear torque multiplication of an ordinary gearbox As the secondary speed increases the torque multiplication factor diminishes, and when it is about equal to unity, disc A is released and disc K is gripped. This establishes direct drive and allows member C to come to rest. Due to the provision of a freewheel F between the sleeve E, carrying the secondary member and the extension piece R of shaft S, the secondary member also is free to come to rest. Thus with disc K gripped the drive is perfectly free, there being no layshaft to idle as in an ordinary gearbox and no churning action of blade members.

Performance characteristics.

The performance of an 85-H. P. bus equipped with the Lysholm-Smith converter is shown in figure 2. It will be seen that the engine r. p. m. increases from about 1 530 to 1 950 while the road speed increases from 0 to 23.4 m. p. h., the rise being practically uniform. The slip in the converter at 23.4 m. p. h. is about 27 %, and the efficiency while the converter is in action follows a curve very similar to that obtainable with electric transmission. It is zero to begin with, as with every kind of transmission, and it touches 86 %. It is over 70 % between 5 m. p. h. (when the torque multiplication factor is about 4 1/2) and 23.4 m. p. h. (when the torque multiplication factor is unity). For a turbine machine operating over a wide range of speed this efficiency characteristic is unusually favourable, the ordinary turbine characteristic falling away rapidly on either side of the maximum.

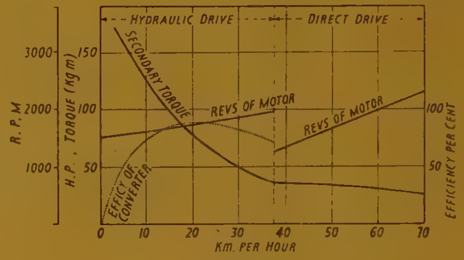


Fig. 2.

The low efficiency of ordinary turbines at any speed other than the optimum is due to the incorrectness of the angles at which the fluid strikes the blading. This incorrectness results in eddying, as shown to the left of figure 3. By rounding the inlet edges of the blading as shown on the right this eddying can be prevented, and it is in this way that a reasonably high average efficiency has been secured with the Lysholm-Smith converter. It will be remembered that in the Vickers-Coats Converter eddying is prevented by pivoting certain blades and thus ensuring that they present themselves to the fluid at the correct angle.

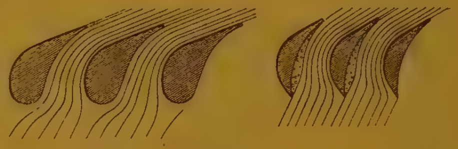


Fig. 3.

The efficiency of the Lysholm-Smith converter is about the same as that of electric transmission, but, owing to the possibility of driving direct when the vehicle has been started, the average running efficiency must be considerably higher. This transmission is also very much lighter and cheaper, and more compact than electric transmission.

Further details.

Hydraulic and electrical machines suffer from the disadvantage that during long periods under heavy load conditions they develop a considerable amount of heat. In the same way that dynamos and motors are provided with fans, the Lysholm-Smith converter is provided with a small oil cooler or radiator. For the reason that the continuous operation of the converter over long periods is necessary only on a stiff gradient the radiator has little to do, and normally the converter keeps very cool. The flow of oil through the radiator is promoted by a pressure difference set up by the pump member, connections being made by tapping the converter casing at suitable points. Across similar points are connected two ejectors, one for raising leakage oil to a gravity tank, and the other for forcing it back into the converter. A connection from the converter casing to the gravity tank is effected through a special leak, to allow the oil in the converter to expand and also to pass away any air or gas which may collect therein.

* * *

The Vickers-Coats hydraulic torque converter.

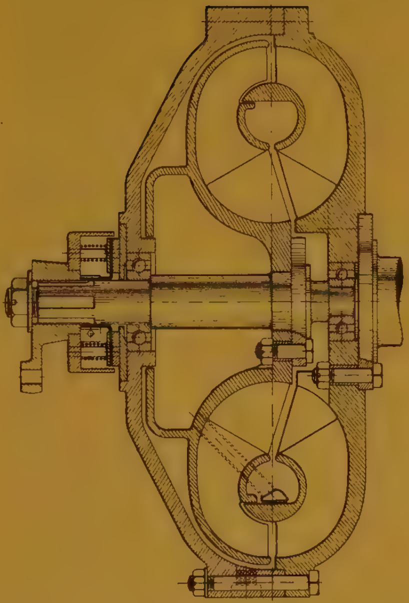


Fig. 4. — The Vickers-Coats hydraulic torque converter.

Comparable with electric transmission for efficiency, but very much simpler and cheaper in construction, is the Vickers-Coats hydraulic torque converter. So far, this has not been supplied except for small powers, but there is no reason why larger units should not be made. A section of this converter is shown in figure 4. It consists of a vaned impeller A, carried round by the engine, and forming the casing for the whole; a vaned impulse receiving member B, geared to the locomotive driving wheels; and a stationary reaction member C, also having vanes. The vanes of the member C are made up of a number of pivoted segments which are free to assume the direction taken by the fluid leaving B. Eddying losses are reduced by giving the fixed vanes this freedom, and it is in this pivoting of the vanes that the Vickers-Coats converter differs from the early Föttinger transmitters. The latter were suitable for one speed ratio only. Figure 5 shows some curves taken from *Engineering* relating to the performance of a converter of this type. It will be

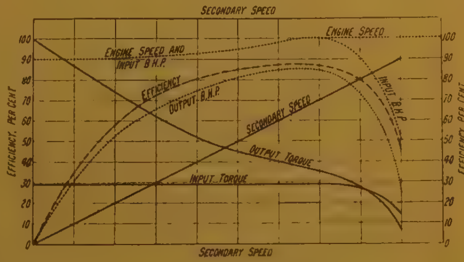


Fig. 5. — Performance curves for Vickers-Coats torque converter.

seen that the efficiency curve, though never quite touching the 88 % mark of the early fixed-ratio converters, is a broad-topped one giving, on the average, a slightly higher efficiency than that obtainable with electric transmission.

The Vickers-Coats hydraulic torque converter has been applied successfully to an internal-combustion locomotive having a two-cylinder airless-injection engine developing 26 B. H. P. at 1 200 r. p. m. This engine is called upon to propel 30 tons in all over the miniature railway at Scarborough, so that the weight/power ratio is high enough here to impose a severe test on the transmission.

A clutch can be incorporated if desired within the Vickers-Coats converter, to give a straight through or 1 to 1 drive under normal running conditions. It should be noted, however, that unless the fluid be withdrawn from the coupling, when a straight through connection is made, the efficiency must be appreciably below 100 %, as the stationary member C will prevent the fluid from being carried round *en bloc,* and will cause eddying losses.

The conversion ratio obtainable with the Vickers-Coats converter is about 3 1/2 to 1, and this, with a fixed reduction gear to suit the power/weight ratio in question, should be quite sufficient for many railway purposes.

The drive is particularly smooth, as there are no gear-changing pauses, and the clutchless converter cannot transmit heavy shocks or torsional vibrations.

* * *

The Vulcan-Sinclair hydraulic coupling.

The drive which is most economical of all from the power loss point of view is the well-known change-speed gear drive. The weakest of several weak links in the conventional automobile drive is the slipping friction clutch which, in a railway application, can be burned out

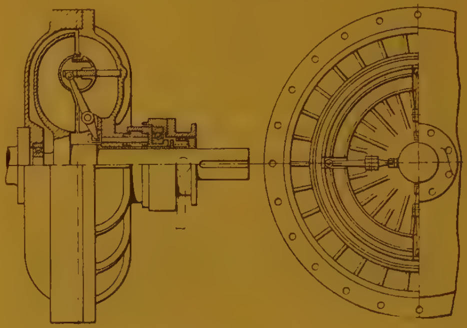


Fig. 6. — Vulcan-Sinclair hydraulic coupling, ring-valve type.

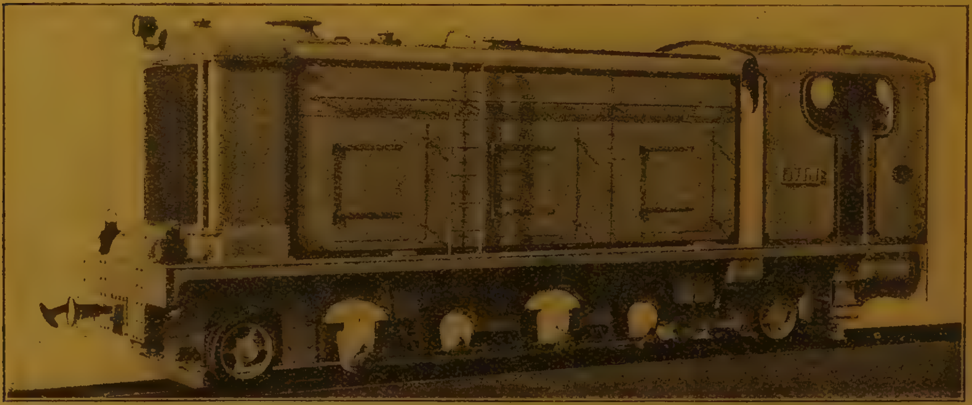


Fig. 7. — Hudswell Clarke Diesel locomotive with Vulcan-Sinclair hydraulic coupling.

completely, and almost at once, unless an exceptionally large and well-ventilated unit is employed. Locomotives up to about 150 H. P., and incorporating multiple plate fabric-lined clutches, are available, and are quite reliable, but for higher powers, and indeed for all powers, the most satisfactory slipping member to use with a gear drive is the Vulcan-Sinclair hydraulic coupling. Known in automobile circles as the « fluid flywheel », this coupling has now been developed to meet the special requirements of railway work.

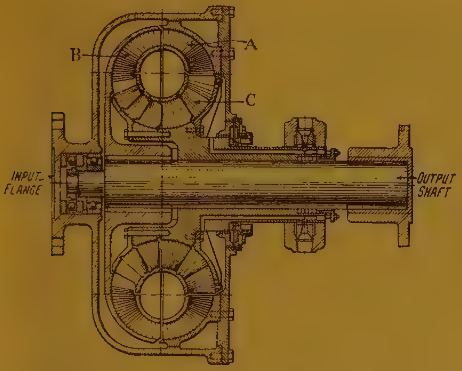


Fig. 8. — Hydraulic coupling with automatic action.

As originally devised for high-power marine drives, and as used to-day in large ships and in automobiles, the Vulcan hydraulic coupling consists of two circular and similar vaned members, these being equivalent to the driving and the driven members of a clutch. The vanes are plane and radial, and are like internal strengthening webs to a dished casting. Without the webs the casting would form a circular trough of semi-oval cross-section, the outer edge and the inner edge being in the same plane. Between the two edges of this larger trough there is another and a smaller circular trough, of semi-circular cross-section and supported by the vanes. The edges of this smaller trough lie in the same plane as the edges of the enveloping trough. In the working assembly the vaned members are face to face, with their corresponding trough edges almost touching, and they rotate about a common axis. The space between them is about 3/16 inch for a 300-H. P. locomotive coupling, and the active fluid, but for the intersection of the vanes, would form an unbroken circular ring of oval circular-cored cross-section. The impelling member carries a cover to enclose the whole device, and the shaft of the follower passes through a bearing in the impeller assembly, there being a simple gland at this point. The gland permits relative motion between the impeller and the follower, but as this relative motion is, under normal conditions, only about 2 % of the engine revolutions, the duties are very light. The pressure to be resisted is very low except at starting, when it may reach 20 or 30 lb. per sq. inch.

Oil is used as the working fluid on account of its lubricating and non-corrosive properties. The thinner the oil, the more efficient the drive, for — and this should be clearly understood — the coupling transmits engine torque by virtue of inertia effects, and not by any viscosity effect. Normally, the impeller runs a little faster (about 2 % faster) than the follower, so that the centrifugal tendency of the fluid is greater in the former member, and this causes the fluid to assume a vortex ring motion, the inner particles passing outwards in the impeller, and the outer particles passing inwards in the follower. The angular velocity of the fluid in the impeller cavities being constant about the coupling axis, it follows that the fluid particles gather fresh momentum and kinetic energy as they move outwards to travel round in a larger circle. This energy, which must be supplied by the engine, is given up to the follower, for here again the fluid particles must move with uniform angular velocity about the coupling axis, but inwards. The slight change in the angular velocity of the fluid particles, as they pass from the impeller to the follower, and back gain, may cause a little eddying, but full engine torque is transmitted from impeller to follower, so that the power loss is proportional to the slip, and is very small — about 2 % in a normal coupling. The heating of the Vulcan-Sinclair hydraulic coupling is far less than in any other hydraulic transmission device, and no radiators, ribs or other cooling devices are required.

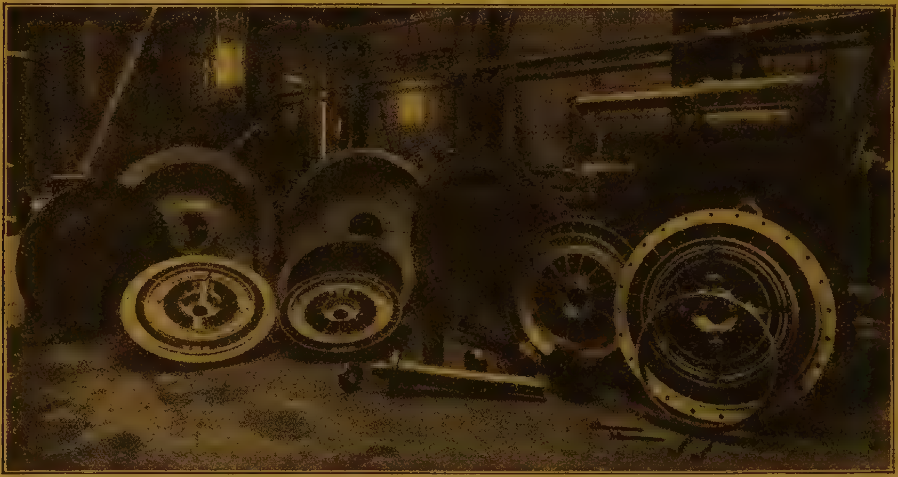


Fig. 9. — Vulcan-Sinclair hydraulic couplings in the works of the Hydraulic Coupling & Engineering Company.

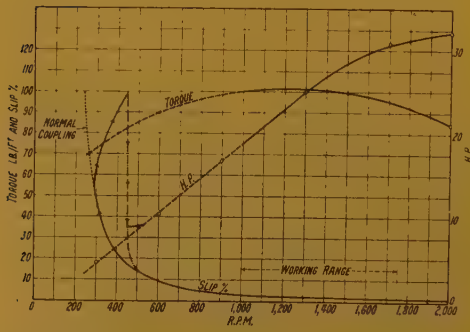


Fig. 10. — Comparison of characteristics of normal and new automatic hydraulic coupling.

The simple Vulcan coupling, from which the Vulcan-Sinclair coupling has been developed, is employed extensively at sea as a hydraulic clutch for connecting marine Diesel engines to geared propeller shafts. It is also used in conjunction with exhaust steam turbines operating on the Bauer-Wach principle. In these applications it prevents the transmission of torsional vibrations and minimises the shocks experienced when, in a rough sea, the propellers are alternately lifted clear of the water and plunged deep therein. About 600 000 H.P. is transmitted through the hydraulic couplings in marine service, and here individual couplings are called upon to handle very large powers. Vulcan-Sinclair hydraulic couplings are widely used in industrial applications, particularly for high-power fan and centrifugal pump drives, and now the Daimler and Lanchester companies use these couplings in conjunction with Wilson four-speed epicyclic gearboxes for their high-class automobiles.

Vulcan-Sinclair hydraulic couplings for locomotives.

The application of a simple Vulcan coupling, or fluid flywheel, to road and rail traction calls for a compromise in design. Thus, a small coupling is desirable to give the necessary slip when starting, otherwise the engine tends to stall when accelerating. But, under normal running conditions, the slip should be as small as possible, which in turn necessitates a larger size of coupling. For automobile work a reasonable compromise is possible, because a full-load slip of 5 % or so has been found by careful fuel-consumption tests to be of no importance. For locomotive work, however, the powers are high, and at times full torque must be transmitted for extended periods; hence it was felt that a 5 % loss could not be tolerated. Steps were taken to make a normal coupling, having a slip of 1 1/2 to 2 %, give the required « free engine » effect, and figure 6 shows the coupling which eventually fulfilled the necessary conditions. A coupling of this improved type is utilised in the 300-H. P. Diesel locomotive shown in figure 7. Here the fluid circulation can be impeded at the will of the driver by means of an externally operable ring valve, and in this way the torque at the follower can be reduced, when required, to a sufficiently low value.

An entirely automatic action is secured in the latest traction coupling shown in figure 8. A reservoir chamber is carried by the runner, or driven half, and into this the working fluid is carried by several scoop-ended tubes (situated between the vaned members) whenever the speed falls below a certain predetermined limit. In this way a sufficiently high slip is secured for idling and starting conditions. At normal speeds centrifugal action ensures complete evacuation of the reservoir and a full coupling. Perfect balance between the centrifugal effect and the « scoop tube » effect is possible at only one speed, and the equilibrium is intentionally unstable so that with the least excess of speed the coupling is definitely full, and with the least fall in speed the working circuit is sufficiently emptied to give the required additional slip. Right down to the critical speed the coupling will deliver full engine torque and sustain the drive. The transition at the critical speed takes place very rapidly, and below the critical speed the transmitted torque is so reduced that the rotation of the runner may be arrested by the application of a « clutch stop » or brake of suitable dimensions.

The characteristic of the new traction coupling is compared in figure 10 with that of the simple coupling, and it will be seen that the new coupling possesses very distinct advantages. It empties when the speed falls below 300 r. p. m., and it does not fill again unless the speed is raised above 450 r. p. m. Over the working range of 1 000 r. p. m. to 1 750 r. p. m. the slip averages 2 % and the efficiency 98 %.

[628. 234 (.73)]

General Electric air conditioning for Pennsylvania diner,

by T. F. PERKINSON and F. L. SAHLMANN,
Transportation Engineering Department, General Electric Company.

(*Railway Age.*)

The Pennsylvania Railroad has placed in service between New York and Washington a diner equipped with a mechanical air-conditioning system manufactured by the General Electric Company. This equipment is of the conventional type employing a direct-current motor-driven compressor, but it incorporates a number of novel features pertaining particularly to the power supply and the arrangement of the compressor and condenser units. It has a refrigerating capacity of 5 1/2 tons of ice per day (66 000 B. T. U. per hour) and uses Freon as the refrigerant.

Another General Electric air-conditioning equipment previously put in operation on this railroad, in suburban service on the Paoli line, is of the hermetically sealed type, taking power from a 25-cycle trolley.

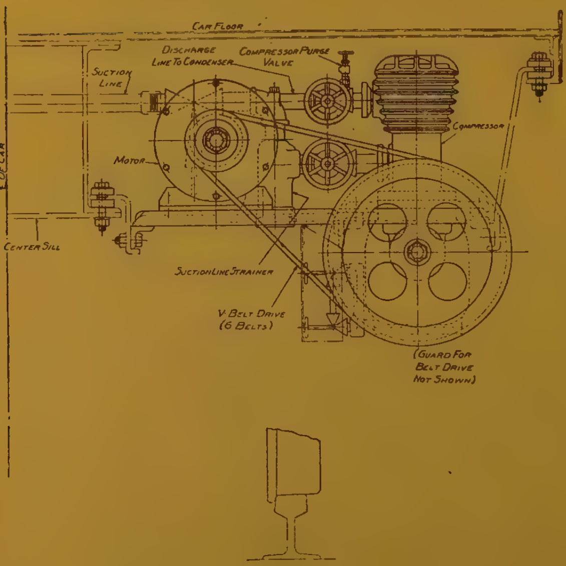


Fig. 1. — Mounting arrangement of compressor unit.

System complete and self-sustaining.

The equipment of the new conditioner consists essentially of one compressor unit located under the car body, one condenser unit located under the car body, two evaporator units located in the clerestory, one 20-kw. axle-mounted generator, and one control equipment.

The compressor, with the necessary valves, fittings, gages, etc., and its direct-current driving motor, are mounted in a framework suspended from the car underframe. The condenser, liquid receiver, condenser fan and fan motor are assembled in a framework independently of the compressor unit, and are also mounted beneath the car body. The air is drawn from the side of the car through the condenser, the circulation being such that no additional space is required at either end of the unit to secure an adequate flow of air irrespective of the direction of the car movement. Each or the two air-conditioning units in the clerestory of the car consists of an evaporator coil, a steam heat coil, expansion valves, circulating fans and fan motor mounted in a framework with the necessary ducts and piping. The equipment is suitable for operation with or without ducts. In this particular installation no ducts are used, the cooled air being discharged horizontally from a grilled opening in the bulkhead at each end of the car.

All-weather control.

The control consists essentially of a differential thermostat, the necessary contactors, resistors, etc., for controlling the operation of the various motors. The differential thermostat has been developed particularly for this application. It operates in such a manner that the outside air temperature modifies the action of the indoor thermostat to maintain the proper indoor temperature. This conforms in general to the practice recommended by the American Society of Heating and Ventilating Engineers. The thermostat controls the power supply to the compressor motor, which is either operating at full capacity or is shut down. The evaporator fans may be operated independently of the compressor unit. This permits operation of the air circulating fans for ventilation even though air conditioning is not required, and also permits use of this system for supplying make-up air during the winter season. Cold drafts are avoided during the heating season by tempering the air by means of a steam heating coil in the evaporator unit.

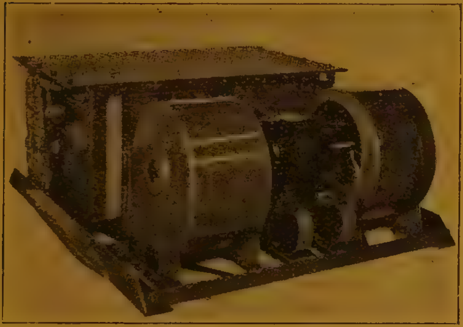


Fig. 2. — Evaporator fan-unit assembly.

The total volume of air circulated is approximately 2 000 cubic feet per minute, a portion of which is make-up air drawn from openings in the side of the clerestory through filters by the fans and forced through the heating and cooling coils into the car. The air from the car returns to the fans through recirculating air openings in the ceiling near the blower set at each end of the car. With a system of this type the interior of the car is maintained at a pressure slightly above atmospheric. Because the incoming air is filtered, there is a noticeable absence of cinders and dust, since the movement of the air is outward from the car through cracks and door frames, around the windows, etc., instead of inward as in the case with the ordinary car ventilating systems.

20-kw. gear-driven generator.

Of particular interest is the source of power supply. An axle-mounted, 20-kilowatt, 37 1/2-volt, direct-current shunt-wound generator, gear-driven from the inside axle of one of the car trucks, provides power for air conditioning, battery charging and lighting, and replaces the conventional car-lighting belt-driven machine.

The generator has been designed to obtain the maximum capacity in the extremely limited space available in the truck. Fabricated construction, involving no castings of any kind in the generator frame, has been employed. The magnetic circuit is made up of rolled steel plate, the magnetic qualities of which permit working at higher flux densities than does the conventional steel casting, and with minimum physical size and weight.

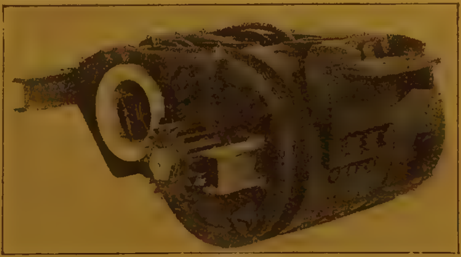


Fig. 3. — The 20-kw. axle-driven generator.

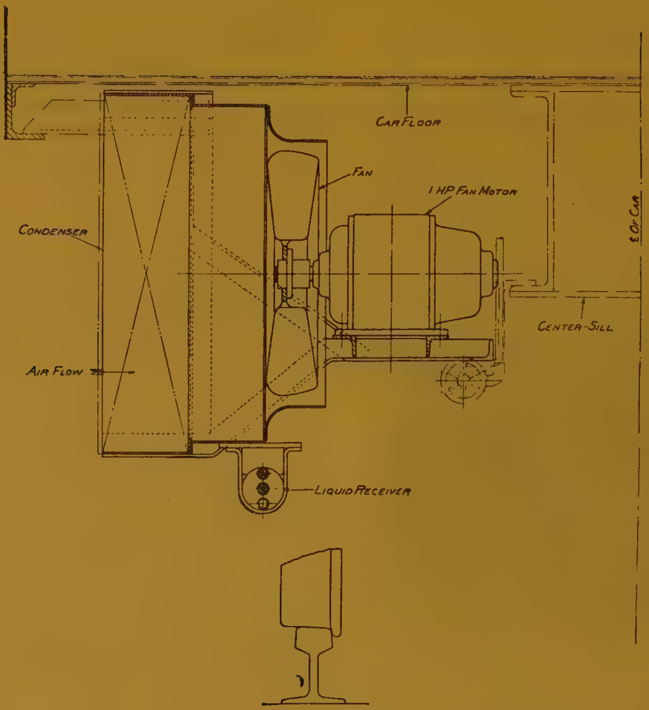


Fig. 4. Mounting arrangement of condenser unit.

A special axle to receive the generator and gear replaces the standard car axle. The axle bearings, of conventional bronze construction, are lubricated with oil and waste, oil being fed from a constant-level oil supply system, identical in principle to the system employed in recent years with General Electric traction motors. Anti-friction bearings are provided for the armature shaft. The

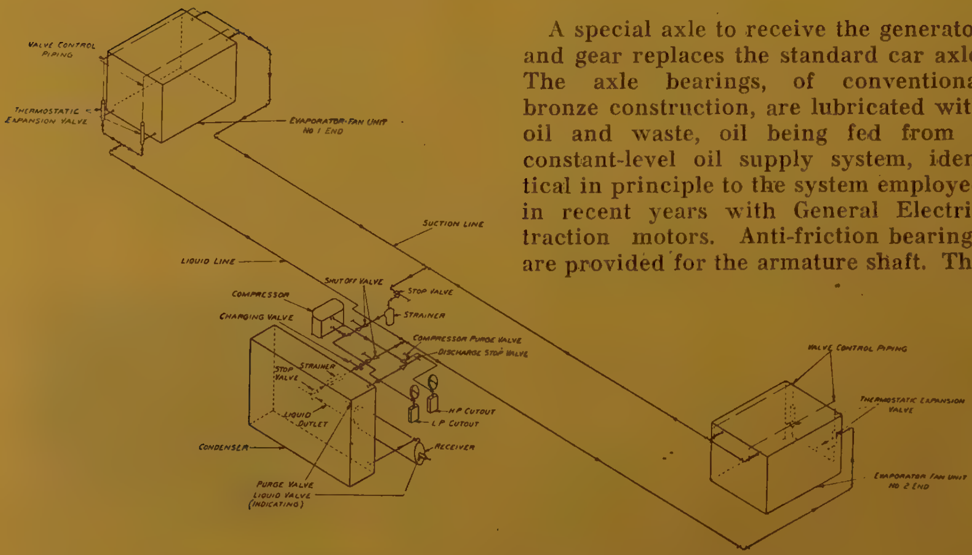


Fig. 5. — Piping layout for complete air-conditioning equipment.

generator has a spring-nose suspension for attachment to the truck frame, with a safety nose as insurance against the generator dropping onto the track in the event of failure of the main suspension.

A textolite pinion is used on the generator armature shaft to insure quiet operation. The pinion and axle gear are enclosed in a welded steel gear case containing the necessary lubricant, and which is readily removable for inspection or maintenance.

The gear mounted on the axle is of the « Safety » type, and is so designed that if for any reason the generator armature should lock or the gearing should jam, the axle is freed from the gear and continues to rotate without developing flat spots on the wheels, necessitating the setting out of the car.

The characteristics of the gear are such that it will not trip free by reason of any coupling shock which the car structure can stand, but will separate if the slipping point of the wheels is approached, as would occur with a locked armature. After an emergency operation the gear can be restored to normal operating status.

Compressor and condenser.

To obtain the maximum of accessibility, the compressor and its driving motor, have been mounted as a separate unit, distinct from the condenser assembly.

The compressor –a single-acting, two-cylinder, relatively slow speed machine running at 400 r.p.m.—is suspended at points near its center of gravity in a simple framework of structural shapes. The design of the framework is such that suspension from the longitudinal outside channel of the car underframe is employed. This arrangement not only provides easy access to all parts of the compressor and its driving motor but also permits addition of a minimum amount of weight in the form of supporting framework.

Drive between the compressor and the 37 1/2-volt, 8-horsepower, direct-current motor is by means of grooved pulleys and endless V-belts, insuring a quiet yet simple and easily maintained drive.

Discharge and suction piping, valves, strainers, high and low pressure cutouts and gages are completely contained and interconnected within the dimensions of the framework, making it necessary to furnish but two refrigerant connections, the discharge and suction lines, to the remainder of the refrigerating system.

The points of support for the compressor unit are insulated from the car body by soft rubber bushings, thus minimizing the transmission of compressor vibration to the car structure.

The space available for the application of compressor and condenser equipment beneath the car underframe is, in the majority of existing rolling stock, very restricted. In particular is this restriction present in the case of dining cars, where this space is utilized to a large extent by provision boxes, battery boxes, brake equipment, etc. Very often the space available for the application of the compressor and condenser equipment is hemmed in on each side by existing boxes — a condition that aggravates the problem of obtaining adequate accessibility. In the installation of equipment for the Pennsylvania diner, the separation of the condenser unit from the compressor and its driving motor facilitates the problem of securing adequate location and accessibility to apparatus.

As with the compressor mounting, a minimum of framework is required to carry the condenser, its cooling fan, and the driving motor for the latter. Easy access to the fan motor for inspection and maintenance is obtained from beneath and behind the condenser.

The condenser fan, a 22-inch four-blade unit of special design for high efficiency and quietness of operation, is driven at 1 200 r.p.m. by a one-horsepower, 37 1/2-volt, totally enclosed, direct-connected, series-wound motor. The fan blades are of cast aluminium with non-resonant characteristics, a factor contributing largely to quietness in operation.

Evaporator units provide year-round service.

There are two evaporator units, one mounted in the clerestory space behind the bulkhead separating the dining salon from the galley, and one in the clerestory at the opposite end of the dining space. Each unit includes two sets of copper coils enclosed in a copper-fin assembly. One set of coils carries the refrigerant gas and furnishes the refrigerating effect for cooling the incoming air. The remaining set is connected to the steam heating system of the car and is employed during periods when heating rather than cooling is desired. Two thermostatic expansion valves are used on each unit.

A 1/3-horsepower, 37 1/2-volt, series-wound motor drives two overhung centrifugal fans which supply 1 000 cubic feet of air per minute to each evaporator unit. The motor is mounted on a spring base which effectively prevents the transmission of vibrations and consequent noise to the car structure.

Fig. 6. — Control group in mounting cabinet with cover removed.

The conditioned air, as it leaves the evaporators, is assisted in its passage to the various parts of the dining salon by two small motor-driven deep-pitch bulkhead fans, comparable in size to the conventional type of railway car ventilating fans. They are of the three-blade type, not unlike a three-blade marine propeller in appearance, and are designed for maximum quietness and efficiency. One fan is mounted directly beneath each of the cold-air outlets and is tilted up to maintain the elevation of the cold-air stream discharge from the evaporator outlet grille.

Pole change and generator regulator.

A novel feature of the generator control equipment is the speed and directional switch, developed especially for this application. This performs the function of securing correct polarity for generator excitation regardless of the direction of operation of the car, without battery drain at standstill.

The use of the speed and directional switch eliminates the necessity of a separate generator exciter or bulky brush-rocking mechanisms, thus permitting full utilization of the available space in the truck for power-producing purposes. Positive build-up of correct polarity is assured by battery excitation—flashing — of the generator field during the accelerating period. When the reverse current relay closes, the generator is self-excited. When the train speed is below approximately 10 miles an hour all excitation is removed by this switch, thus avoiding battery drain.

A carbon-pile regulator holds the generator voltage within a narrow voltage range between car speeds of 25 and 90 miles an hour. A current element on this regulator limits the current output to a safe value.

The generator and compressor motor control equipment, with the exception of the carbon-pile regulator and reverse current relay, is in a sheet metal box suspended beneath the car body adjacent to the condenser unit. The carbon-pile regulator and reverse current relay are designed to replace corresponding lighting generator regulator equipment commonly found in locker space on existing passenger-car equipment.

High and low pressure cutouts, mounted with the compressor unit, protect the refrigerant system from excessive and subnormal pressures by shutting down the compressor should abnormal conditions hold.

[621. 153.4 & 621. 153.5]

Soo line tests cyclone front end.

(Railway Age.)

The Minneapolis, St. Paul & Sault Ste. Marie has recently completed some interesting tests of the locomotive front-end arrangement, known as the Cyclone front end and described on page 813 of the *Railway Age* issue of 25 April, 1931. These tests indicate that the device functions effectively not only to prevent the emission of sparks from the smokestack but to permit operating with an appreciably larger exhaust nozzle, reduced back pressure and generally improved locomotive performance as regards hauling capacity, speed and fuel economy. The use of the Cyclone front end, as installed on the Soo Line test locomotive, permitted increasing the exhaust-nozzle diameter 9/16 inch and reducing the back pressure approximately 4 lb., or 27 to 50 %, dependent upon the speed and cut-off. The locomotive steamed freely, maintaining an average superheated steam temperature of 636° F. with a front-end gas temperature of 570° F.

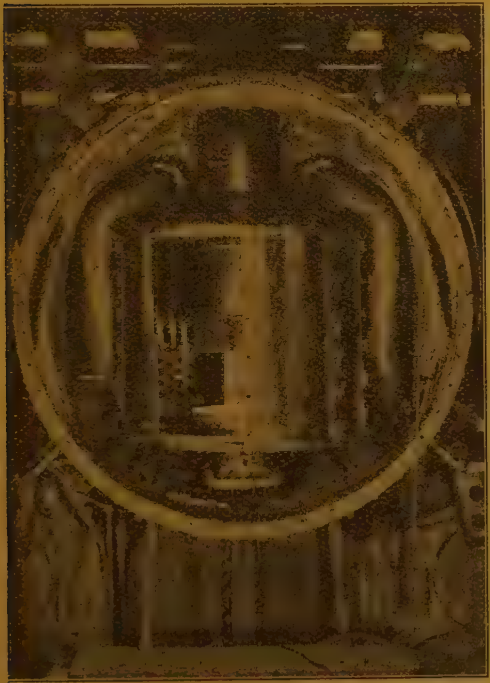


Fig. 1. — Cyclone front end as installed with one front section removed to show the interior arrangement.

The Cyclone front end was originally developed on the Northern Pacific and subsequently applied to several hundred locomotives in order to permit burning sub-bituminous coals of the Rosebud variety without spark emission and attendant fire hazard. As described in the article referred to, the device consists essentially of a sheet-metal drum equipped with breaker strips around the interior circumference and deflecting vanes at the intake, so arranged that exhaust gases are given a centrifugal motion which delays the cinders in the drum long enough for them to be reduced in size and extinguished before being

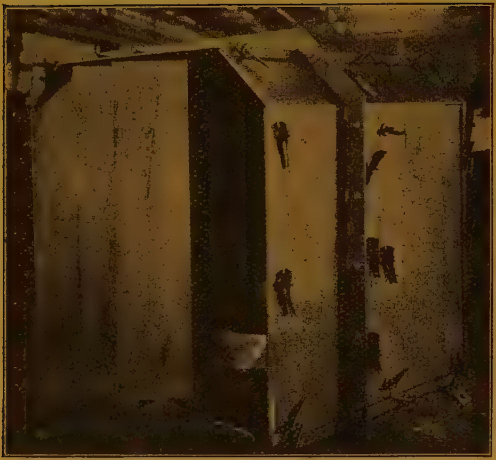


Fig. 2. — Spark-arrester drum before installation in the locomotive. — View taken from the intake end.

discharged to the atmosphere. No netting or similar restriction to draft is required with this construction.

The initial Cyclone front-end design, now called the Type A, was installed with the gas intake at the front of the smokebox and usually necessitated an offset exhaust nozzle. After manufacturing rights were acquired, in 1931, by the Locomotive Firebox Company, Chicago, the improved Type-B Cyclone front end was developed, being essentially like the earlier type, except for the omission of the top head of the drum and the utilization of this space to give greatly increased spark-arrester drum volume and a larger front intake area. Moreover, it was not necessary to use an offset exhaust nozzle with this design. A still further improvement was effected with the Type-C front end used in the Soo Line tests, whereby the intake to the drum was reversed and placed directly in front of the flue sheet, thus providing a much shorter and less restricted path for the flue gases than with either of the preceding types or with the conventional Master Mechanics' front-end arrangement.

The general arrangement of the Type-C front end is illustrated in figure 1. It surrounds an inside stack-extension and is used in connection with a low, straight exhaust base. The spark-arrester drum is manufactured in segments, keybolted together so that it can be applied either in a single unit or dismantled, and applied section by section It can be removed in the same way. It is manufactured from 1/4-inch sheet steel and has a heat-treated steel liner 1/4 inch thick carrying vertical breaker strips applied to the inside surface. This liner, arranged in segments to correspond with the outer wall, is spotwelded to the walls of the drum and can be readily removed. The liner is used to prevent wear to the walls of the drum itself. A vertical baffle plate, surrounding the opening into the drum, shuts off from the draft area all of the smokebox, from the superheater header forward. This arrangement excludes all steam pipes and joints, with the exception of the superheater units, from the draft area and, therefore, minimizes the possibility of steam leaks or outside air leaks around the door ring and steam pipes interfering with the draft.

Method of conducting tests.

In making comparative tests of the new front-end arrangement, the Soo Line decided to take a locomotive in regular service and conduct a test first with the conventional Master Mechanics' front-end arangement and then with the Cyclone Type-C design. A *Mikado* locomotive, with 28-inch by 30-inch cylinders and 170 lb. boiler pressure, was selected. In preparing for the test, the locomotive was equipped with draft gages in three places, one in the firebox, one at the front flue sheet and one inside the netting in front of the baffle plate. Provision was made to determine cylinder back pressures by means of an Ashton 30-lb. gage in the cab, equipped with a dampener and connected to a 3/4-inch pipe to the smokebox with one branch just below the exhaust-nozzle tip and one to the exhaust end of the valve chamber at the right side. A pyrometer was used with thermocouples applied in the right steam chest and at the front flue sheet to give readings of superheated steam and front-end gas temperatures, respectively, by simply using a multiple-throw switch. The valve gear was calibrated and a scale made which fitted the contour of the reverse quadrant, cut-off positions being read direct by reference to the valve-travel scale. The locomotive was operated with a full throttle opening and speed controlled by cut-off adjustment.

Test runs were made between Schiller Park, Ill., and Fond du Lac, Wis., over a 140-mile division in fast freight service, hauling approximately 1 900 tons. The tests were conducted in the middle of the winter so that this 1 900 tons was approximately 200 tons below the maximum summer rating.

In normal service with the Master Mechanics' front-end arrangement, the nozzle tip used on this class of locomotive is 6 1/8 inches in diameter with no bridge or projection of any sort. This is equal to 29.46 sq. inches of opening. The locomotive was operated on several trips with this nozzle to develop the average back pressure, superheat, and draft under normal operating conditions. Observations taken indicated that the locomotive was a « good steamer » and in first-class condition mechanically, handling the train without difficulty. Back pressures were relatively high, however, and a loss in draft efficiency of 40 % between the nozzle tip and the front of the flue sheet was observed.

When as satisfactory results as possible were obtained with the Master Mechanics' front-end arrangement, the locomotive was tied up and the Cyclone Type-C front end applied. On the first subsequent trip, the exhaust nozzle tip was opened to 6 3/8 inches, giving an area of 31.91 sq. inches, an increase of 8.3 %. One trip was made with this nozzle arrangement, which indicated that the nozzle was still too small, so, for the next trip, the nozzle was opened to 6 5/8 inches, giving an area of 34.47 sq. inches, or an increase of 17 %. The locomotive is reported to have steamed well, possibly better than with the 6 1/8-inch nozzle in the Master Mechanics' front end, but, to insure the maximum possibilities in steaming capacity and general serviceability, the nozzle was reduced to 6 9/16 inches, which was an area of 33.82 sq. inches and an increase of 14.7 % over the 6 1/8-inch nozzle. After it was decided that 6 9/16 inches would be the standard nozzle, several test runs were made and observations taken to determine the back pressure, superheat and draft.

Figure 3, which is a composite graph showing operation over the entire division, indicates the relative back pressures developed at given speeds and cut-offs between the two sizes of exhaust nozzle, or, in other words, between the two different front ends. Referring to the graph, it will be noted that, with the 6 9/16-inch nozzle, the locomotive

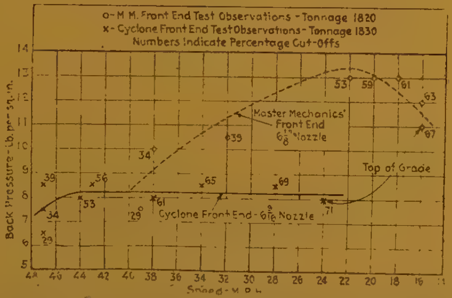


Fig. 3. — Composite graph of several trips showing the average readings at various speeds and cut-offs over a 140-mile division.

developed practically 4 lb. back pressure at 19 % cut-off, this back pressure increasing to a maximum of 10 1/2 lb. as the cut-off was increased to 54 %, then dropping rapidly as the cut-off was further increased and the speed reduced. Under the same conditions, the locomotive, equipped with the Master Mechanics' front end and the 6 1/8-inch exhaust nozzle, developed 8 lb. back pressure at 19 % cut-off, and increased to a maximum of 14 1/2 lb. at 54 % cut-off, then dropping with reduced speed as in the tests with the larger nozzle. An examination of this graph indicates that the locomotive, when equipped with the larger nozzle, developed practically 4 lb. less back pressure throughout the entire normal range of operation, which is equivalent to a reduction of 27 to 50 %, dependent upon the speed and cut-off.

The graph figure 4 shows the relative performances of the locomotive with the two front-end arrangements on the ruling grade, as measured in back pressures, speeds and cut-offs. It is interesting to note that, with 29-% cut-off at the foot of the hill, the Cyclone-equipped locomotive at a speed of 47 m. p. h. developed 6 1/2 lb. back pressure. To maintain speed on the hill, the cut-off was increased to 34 % and again to 39 %, the speed remaining constant to this point. As the increasing grade reduced the train speed, the cut-off was increased at intervals from 53 to 71 % at the top of the grade where the speed was 24 m. p. h. During this time, the back pressure remained practically constant at 8.2 lb., showing that the locomotive had a reserve power capacity which could have been used, if necessary.

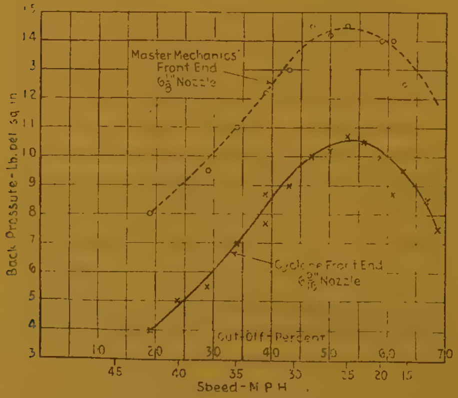


Fig. 4. — Comparative back pressures developed on ruling grade by Soo Line locomotive equipped with the Master Mechanics' and subsequently with the Cyclone front end.

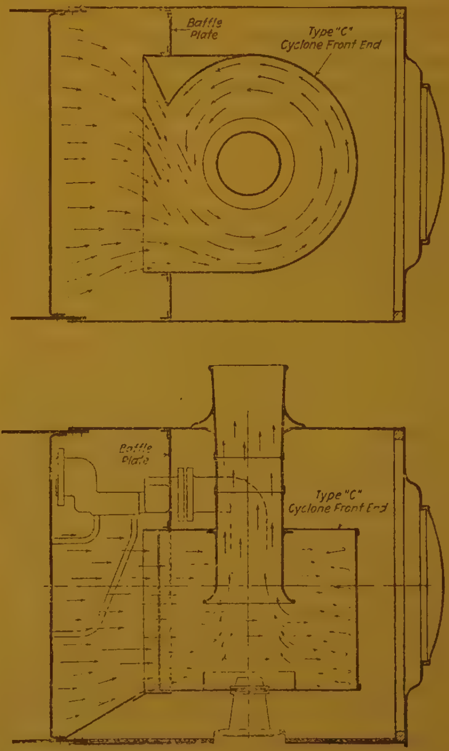


Fig. 5. — Diagram showing gas and cinder travel in the Cyclone type-C frond end as applied to the Soo Line locomotive.

Under the same grade and operating conditions, the locomotive with the Master Mechanics' front end developed a speed of 39 m. p. h. with a cut-off of

29 % and a back pressure of 7 1/2 lb. at the foot of the grade. In an effort to maintain this speed on the grade, the cut-off was increased to 34 %, bringing the back pressure up to 10 lb., the speed falling slightly, however, to 38 m. p. h. As the speed continued to decrease the cut-off was increased at intervals until, at 53 %, a maximum back pressure of 13.4 lb. was developed at a speed of 22 m. p. h. From this point on, both the speed and back pressure dropped rapidly as the cut-off was further increased to 67 % at the top of the grade when the back pressure was 12 lb. and the speed 16 m. p. h. While the tractive force of the locomotive during this test continued to increase until the top of the grade was reached, the maximum horse-power was developed before reaching the top.

While the graphic charts give a visual picture of the definite improvement in locomotive performance, due to the application of the Cyclone front end, with the resulting increase in nozzle area, an additional advantage was observed in the increased smoothness of operation of the locomotive after the nozzle tip had been enlarged. In every instance, where it was necessary to work at rather high capacities, the Cyclone-equipped engine was said to work much easier, and be capable of handling increased tonnage. Less grate shaking was required and a thinner fire kept at all times, burning the same kind of fuel.

The results of these tests with the Cyclone front end, covering a period of about three months, indicate that by the use of this device, the following advantages may be anticipated: Larger nozzle area, decreased back pressure, better steaming, reduced fuel consumption, sparkless operation, no interference with draft from air leaks into the smokebox, and increased hauling capacity.

[621. 335 & 621. 43]

High-power Sulzer Diesel locomotives for express passenger and for goods trains.

(*Le Génie Civil.*)

The application of the Diesel motor to railway traction is developing rapidly, as the *Génie Civil* has shown in an article specially devoted to this subject in the number for December 31st, 1932. At the present time, this motor is being applied more especially to the three following types of locomotive:

a) Shunting locomotives, the power being, according to the case, 300 to 330 H. P. (U. S. A., Rosario and Buenos Ayres ports), 600 H. P. (U. S. A. and France), and 750 to 1 000 H. P. (Manchuria, France, U. S. A.);

b) Rail motor coaches and locomotives for secondary lines and branches of main lines; their power is between 150 and 450 H. P.;

c) Locomotives of average power which so far are only in regular working in the Argentine (locomotives and mobile power stations of 1 200 to 1 500 H. P. on the Buenos Ayres Southern Railway) and in Siam (450 to 900 H. P.).

There are certain technical difficulties in the way of the construction of large Diesel locomotives and especially as regards the possibility of obtaining a sufficiently low weight per H. P. Important progress has been made in this direction, however.

Many railways on which the cost of

steam traction proves burdensome, may be expected ultimately to prefer Diesel locomotive traction, with its important advantages, to electrification.

Generally speaking, the cost of motive power is a maximum with steam traction and a minimum with electric traction obtaining its supply from hydro-electric stations. On the other hand, the first cost is lowest with steam traction and highest with electric traction. The other costs are nearly proportional to the density of the traffic. Diesel traction occupies a mid position. When the traffic is light, the Diesel is a very serious competitor to steam traction, whilst in the case of lines with very heavy traffic, electric traction has shown itself to be the best. It, therefore, seems that Diesel traction is, in principle, the best economically speaking for average traffic.

There is, moreover, a large number of cases in which economy is not the only question to be considered. In regions where water is scarce and fuel has to be conveyed long distances, the Diesel motor is obviously favoured by local circumstances. The same applies to strategic lines.

Since the War, most railway companies have put into service locomotives of steadily growing power. This development did not present any great difficulties in the case of the electric locomotives, thanks to it being possible to drive each axle independently, but was less easy in the case of the steam locomotives, because of the difficulty of getting a sufficiently large boiler within the available gauge limits and through having to couple together the driving wheels. It is most important for the future of the Diesel locomotive that it should reach the power of the steam locomotive. The *Revue technique Sulzer* (No. 1 of 1933) reports the investigations of the Sulzer Company in this direction. These studies show that Diesel locomotives of 2 000 to 4 000 H. P. can now be built, which weigh much the same as steam locomotives of equal power.

The whole of the proposed designs include electric transmission which, for powers such as those considered, appears the only practical one at the present time.

The Diesel-electric locomotive for Russia, fitted with two Sulzer motors of a total power of 1 600, built by Messrs. Fr. Krupp, and the 1 700-H. P. Diesel-electric locomotive, built by Messrs. Armstrong Whitworth as general contractors for the Buenos Ayres Southern Railway, are the two largest locomotives with Sulzer-Diesel motor at present in use. The latter of these two locomotives is the largest Diesel locomotive in the world, forming a single unit. These locomotives each have two Diesel motor generator sets, whereas the present type has only a single group like that shown in figure 1, which is a 2 000-H. P. locomotive for passenger trains.

In Europe, where the permanent way is lighter than in America, nose-suspend-

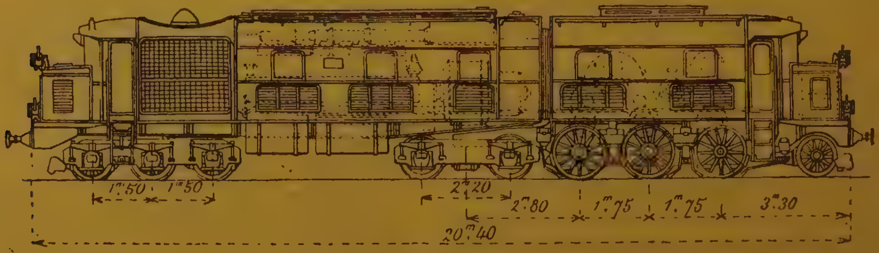


Fig. 1. — 2 000-H.P. Sulzer Diesel-electric locomotive for express trains.

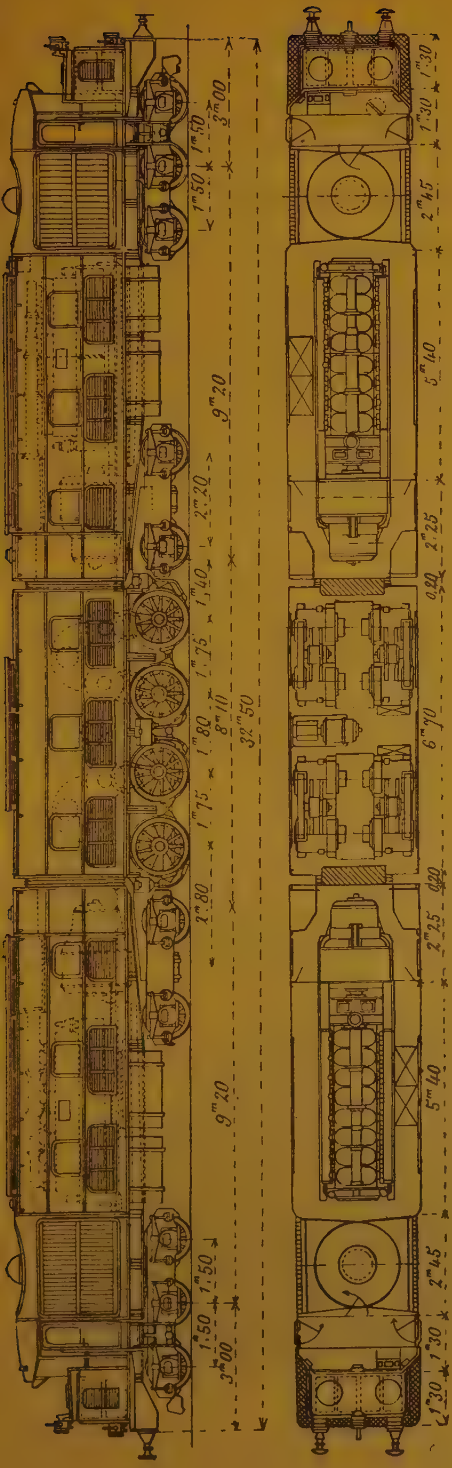


Fig. 2. — 4 000-H. P. Diesel-electric locomotives for express trains.

ed motors are not used for speeds above 100 km. (62 miles) an hour, and for more than 300 H.P. per axle. Consequently the locomotive shown in figure 1 is designed for individual axle drive, the two motors for each axle being carried on the main frame. The power is transmitted through a double set of gears and a quill surrounding the axle. The electric equipment is carried in a separate compartment on a frame independent of that carrying the Diesel motor. If this frame were carried on its own wheels, the whole of its weight would be lost for adhesion, while the other half of the locomotive would not be heavy enough by itself to give sufficient weight on the driving and carrying wheels. In the final design one of the ends of the body in which the Diesel motor is housed, is carried on a bogie, while the other rests on the driving wheel frame.

The Diesel motors of these locomotives consist of two banks of cylinders, each driving a crank shaft. The generator is driven by a gear coupling together the two crank shafts. This arrangement was adopted in preference to the single bank, owing to the lesser height and length required and to the better use made of the available width. A motor of equal power with a single bank of eight cylinders would run at least as fast as the motor with two banks of six or eight cylinders: however, it would require a larger electric generator. In addition, as the cylinders would be much larger, it would be necessary to make provision for cooling the pistons which would necessitate additional cooling.

Locomotives of even greater power are being designed on the same principles. They have a larger number of cylinders and driving axles. The 4 000-H. P. locomotive for express trains (fig. 2) has two Diesel motors carried on two distinct frames separated by a central truck carried on 8 driving wheels and containing the electrical equipment.

The tractive effort at the wheel tread

for which these locomotives have been designed has been calculated from formulæ based on those of Strahl. The value in kgr. is :

$$Z_R = w_L \, . \, G_L + w_A \, G_A,$$

G_L and G_A representing respectively the weight of the locomotive and that of the train behind it, in metric tons; w_L and w_A the corresponding specific resistances in kgr./tons; the values taken for the specific resistances on the straight are :

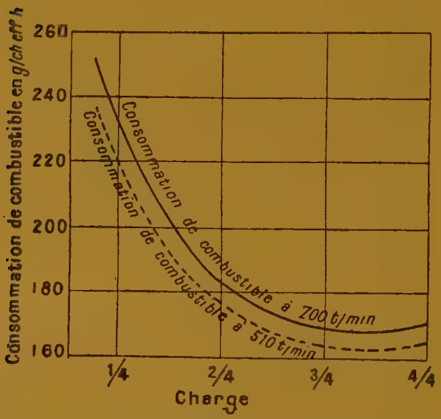


Fig. 3. — Consumption curves of 4-stroke, 8-cylinder, 880 effective H. P. Sulzer Diesel electric locomotive engine.

Note. ▬▬▬ Consumption when running at 700 r. p. m.
——— Consumption when running at 510 r. p. m.
Charge = Load.

a) In the case of express trains :

$$w_L = 3.5 + 0.6 \frac{F}{G_L} \left(\frac{V + 12}{10}\right)^2,$$

$$w_A = 2.5 + \frac{1}{40} \left(\frac{V}{10}\right)^2;$$

b) In the case of goods trains :

$$w_L = 3.5 + \frac{F}{G_L} \left(\frac{V + 12}{10}\right)^2,$$

$$w_A = 2.5 + \frac{1}{20} \left(\frac{V}{10}\right)^2.$$

F is the cross section of the locomotives in square metres and V the speed in kilometres per hour. The value given by W_A in the case of goods trains supposes that the train hauled is composed of empty wagons and loaded open or covered wagons.

Figure 3 shows, in terms of the load, the fuel consumption of a 4-stroke, 8-cylinder motor, with mechanical-injection developing 800 H.P. at 700 r. p. m. These results were obtained during the acceptance trials of a motor for the Chemins de fer de Ceinture, Paris.

[621 .132.3 (.44)]

Two-cylinder simple expansion 4-6-2 express locomotives for the Alsace-Lorraine Railways.

(*The Railway Gazette.*)

As a result of experience gained with a German P8 class 4-6-0 locomotive, fitted with Caprotti valve gear two years ago, two new *Pacific* locomotives have been built on the same principles for the Alsace-Lorraine Railways. The main dimensions of these locomotives are given under the drawings hereafter.

The locomotives embody many interesting features, the principal of which, of course, are the Caprotti poppet valves. The cam boxes and valves are of the latest designs and the driving gear for the cam boxes is of the usual Caprotti standard inside type, which is claimed to possess the great advantage that there is no need to interfere with it when attending to back ends or side rods, that there is a minimum liability to damage and the minimum number of components. The reversing gear is of the same type as that fitted to certain Caprotti geared engines on the German State Railways. The departure from the usual French practice of compounding for express passenger work was due to the contention that with a well-designed poppet valve gear the engines can be worked with early cut-offs and thus an expansion of the steam comparable with compounding can be obtained.

It is understood that a number of special trials with a dynamometer car are to be undertaken with these locomotives. For one of these tests provision has been made in the design to allow special liners to be fitted to the cylinders, whereby the diameter can be increased or de-

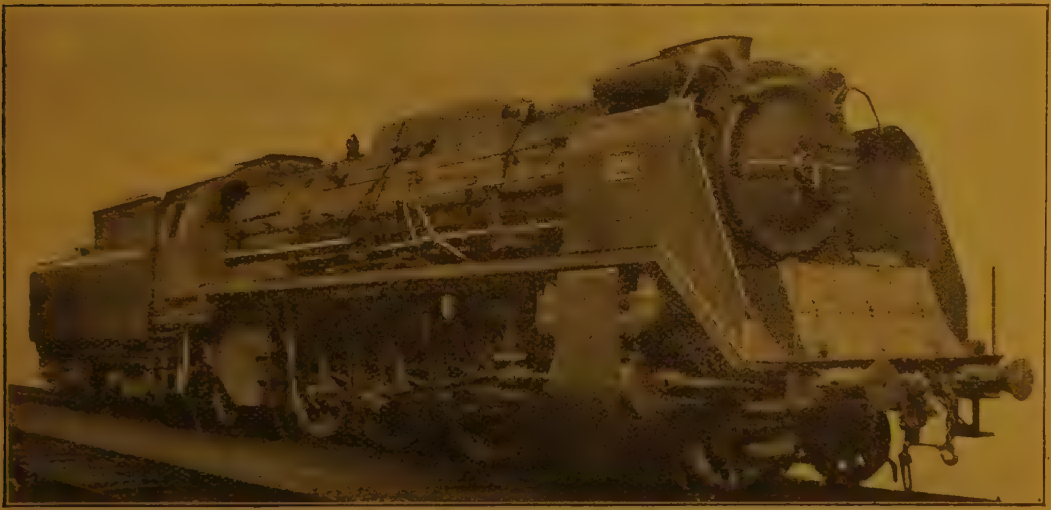
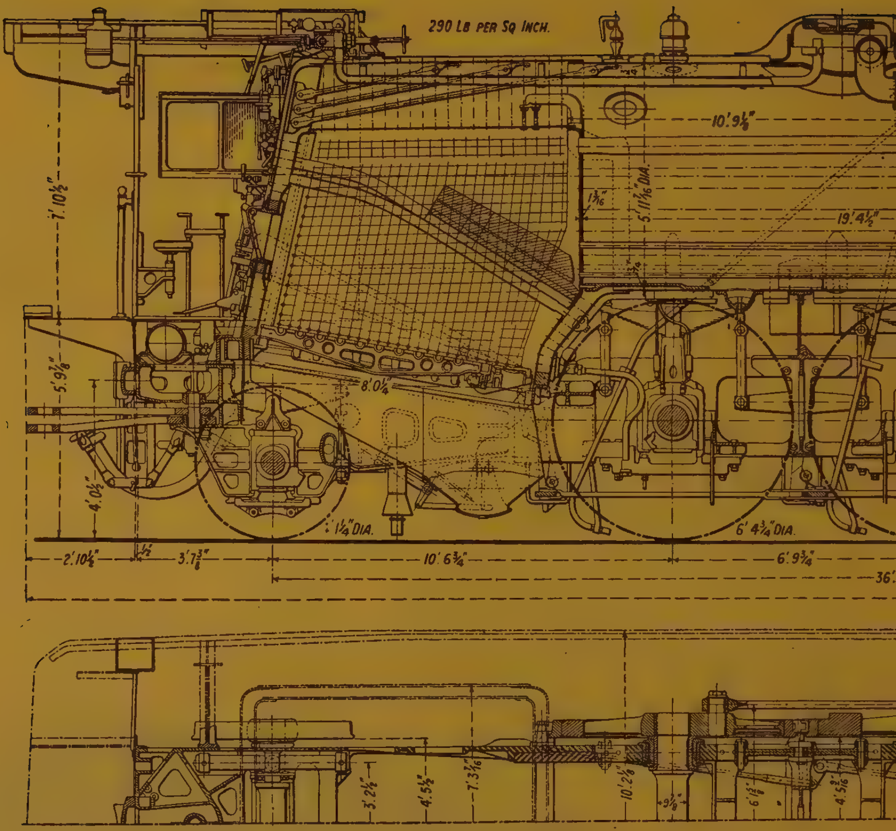
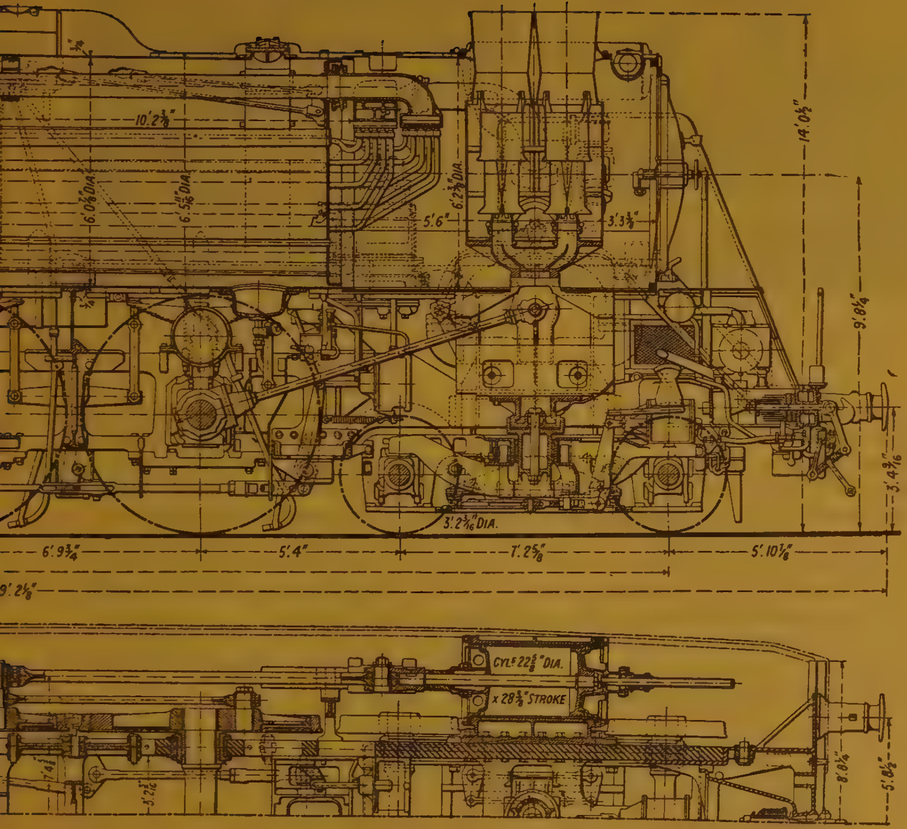


Fig. 1.



Cylinders (2), diameter.	22 5/8 in. (575 mm.).
» stroke	28 3/8 in. (720 mm.).
Wheels, coupled, diameter	6 ft. 4 3/4 in. (1 950 mm.).
» leading bogie, diameter.	3 ft. 2 3/16 in. (970 mm.).
» trailing, diameter.	4 ft. 1 1/4 in. (1 250 mm.).
Wheelbase, coupled	13 ft. 7 1/2 in. (4 150 mm.).
» total engine.	36 ft. 8 7/8 in. (11.195 mm.).
Boiler, working pressure	290 lb. per sq. in. (20 hpz.).
Distance between tube plates	19 ft. 4 1/2 in. (5 905 mm.).
Diameter inside.	6 ft. 0 7/8 in. (1 851 mm.).

Fig. 2. — New simple-expansion 4-6-2 type express locomotive for the Alsace-Lorrain



Heating surface :		
Tubes (30 large and 137 small)	2 205 sq. ft.	(205.0 m^2).
Firebox	201 »	(18.7 »).
Total. . .	2 406 sq. ft.	(223.7 m^2).
Superheater	794 »	(73.8 »).
Grate area	48.5 »	(4.5 »).
Weight of engine in working order	106 tons.	
» » tender » »	66 »	
Total. . .	172 tons.	
Adhesion weight	60 »	
Tractive effort, at 85 % boiler pressure. . .	46 000 lb.	

ys. — Built by the « Société Alsacienne de Constructions Mécaniques », Graffenstaden.

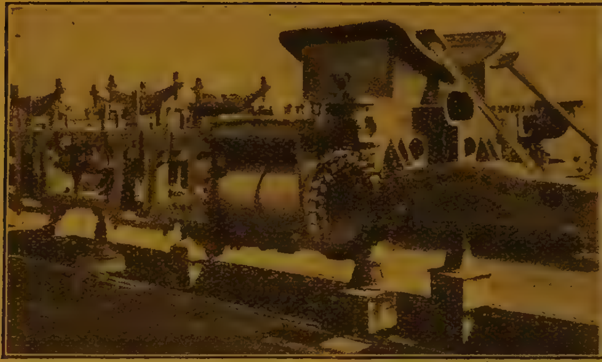


Fig. 3. — Cylinders, smokebox, saddle and springing.

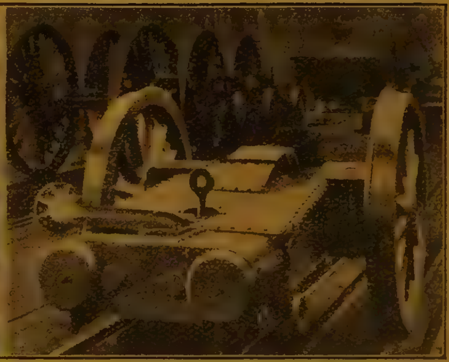


Fig. 4. — Booster truck fitted to one of the engines.

creased at will, the object of such tests being to determine the relation between different cut-offs and the steam consumption per drawbar H.P.-hour at varying outputs of the boiler. The desirability of fitting a booster, in conjunction with the use of a smaller cylinder diameter, will also be investigated during these tests, a booster having been built to fit on to the trailing axle of one of the locomotives.

The new locomotives have bar frames, and the cylinders and smokebox saddle are made in one steel casting, with cast-iron liners fitted in the cylinder barrels. Kylchap double-blast pipes are fitted, and the equipment includes an A.C.F.I. feed-water heater, a live-steam injector and a pneumatically operated firedoor. It will be noticed from the drawing that the footplate is extended over the tender in the same way as the latest express locomotives of the Est and the Etat.

[621.335 (.42) & 621.43 (.42

Diesel-electric streamlined rail coach.

(Engineering.)

The opportunity recently occurred of making the journey from London to Hertford on a Diesel-electric rail coach, which has been designed by Messrs. Sir W. G. Armstrong, Whitworth and Company (Engineers), Limited, Thames House, Westminster, London, S. W., to provide a frequent high-speed local service on either main or branch lines, with the object of assisting the railway companies to compete with motor vehicles running on parallel routes. The coach must be distinguished from the well-known « Tyneside Venturer » coaches of the same firm, since when fitted with a 95 brake horse-power engine, it weighs only 17 1/2 tons in working order *i. e.,* 0.31 ton or 0.25 ton per passenger seat, depending on whether the carrying capacity is 57 or 71 persons.

The coach, of which views appear in figures 1 and 2, is 52 ft. 6 in. long over the body, and has a maximum width of 8 ft. 11 1/2 in. Its height above the rail level is 11 ft. 11 1/4 in., and it is carried on two bogies with a wheel base of 7 feet. The diameter of the wheels is 2 ft. 9 in. and the distance between the bogie centres 30 feet. The maximum axle load, with the car loaded, is 6 tons. As will be seen, the ends and sides of the coach are streamlined, so as to reduce the air resistance to a minimum compatible with the length and width. A deep skirting round the sides is designed to diminish the eddies under the frame, and by reducing the head resistance in this way, it is claimed that a saving of power of no less than 30 % can be effected at normal cruising speed.

The underframe of the car is built up of two pressed-steel, flanged plates, which are fish-bellied, so that unnecessary weight is eliminated. These longitudinals are connected at intervals by pressed-steel stretchers, between which the engine, generator, and motor are slung. A bumper is fitted at each end of the frame and links are provided, so that in emergency the coach can be coupled to standard draw gear. The bogies, which are of the swing bolster type, are unusually light, a result which has been rendered possible by the use of welding in their construction. Their frames are of box-girder form, and were built up of flanged plates and pressed channel sections. The axle-box springs are laminated and are carried upon hangers fitted with Silent Bloc rubber bushes. The vehicle underframe is supported on hemispherical side bearings by rubber pads, the central guiding pivot being free from weight. The wheels are built up on large wooden centres and the tyres are bonded to the axles to allow track circuits to be operated. The axle boxes are of the Isothermos type.

The power unit consists of an Armstrong-Saurer commercial vehicle, six-cylinder, heavy-oil engine, the output of which is either 95 brake horse-power or 140 H.P., and the speed, 2 000 r. p. m. or 1 500 r. p. m. It is arranged with three-point support in rubber Silent-Bloc bushes, while the risk of any vibration being transmitted to the chassis is further prevented by the use of laminated spring suspension. The engine is directly coupled to the generator, which is bolted to the crank case. This generator, which also serves to provide current for lighting, is in turn connected to the traction motor, which is rigidly mounted on the underframe and power is thence transmitted to the road wheels through a cardan shaft and Craven-Guest worm gearing. The electrical equipment was manufactured by Messrs. Laurence Scot and Electromotors, Limited, Norwich. Cooling is effected by two radiators, one at each end of the car, these being mounted in openings in the streamlined skirting.

Control is effected from the driver's cab at each end by three levers. These work in a slotted cover and operate the throttle, the reversing switch and the switch for starting and stopping the engine, respectively. The movements of these levers are transmitted through compensated heavy-duty steel wire ropes running on pulleys to the actual switches, which are situated near the engine. The throttle handle is fitted with a dead-man's button. The brakes consist of Ferrodo-lined expanding shoes, which work on drums mounted on the inside of the running wheels. The four brakes on each bogie are operated from a compressed-air cylinder, which is linked with the brake rigging in the driver's cabs. The rise and fall of the axles in the horn guides is allowed for by suspending the brakes by means of lubricated bushes on the axle. These brakes are applied if the dead-man's button is released.

The body was manufactured by the Park Royal Coachworks, Limited, London, to the designs of Messrs. Armstrong, Whitworth and Company (Engineers), and is an example of how light road-vehicle construction can be adapted to railway service. The oak floor framing is flitched with No. 8 gauge mild steel plate to the pillars, while the rails and other parts are also flitched at all the important joints. The roof is of silver spruce tongued-and-grooved board, and is supported on ash hoopsticks strengthened with mild steel castings. The front and rear portions are, however, of No. 20 gauge steel with a silver finish, which was beaten to shape by hand, the whole being covered with waterproof

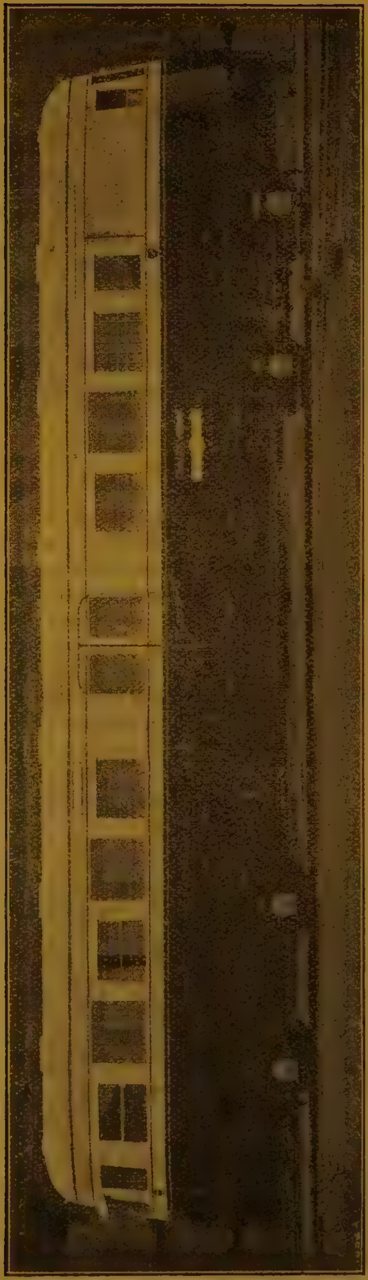


Fig. 1.

canvas coated with white lead. The exterior of the coach is panelled in No. 18 gauge aluminium, excepting the ends where hand-beaten steel is again used. The ceiling is of Salamander panelling, and the interior panelling of birch ply. The seating is of the Park Royal « tub » type. Each seat accommodates either two or three persons and is upholstered in moquette, the framing being of plywood flitched with steel. The floor is of 3/4-inch tongued-and-grooved pine boards, and in the passengers' and driver's compartments is covered with Insulwood, and then with a 3/16-inch cork carpet. Heating is effected by a radiator through which the cooling water from the engine passes, a fan directing the warm air upwards through ducts which run the fulll length of the passenger compartments. These ducts are fitted with louvres, so that the heat is adjustable.

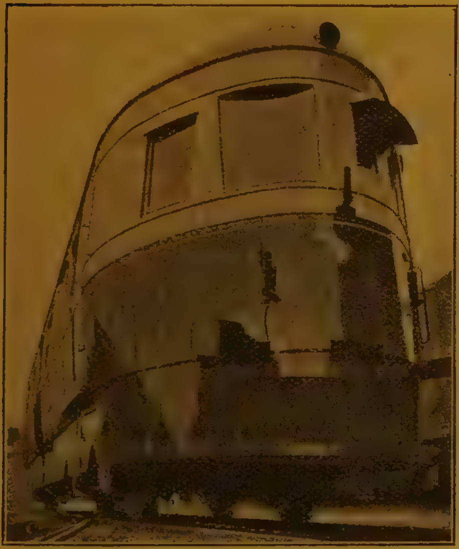


Fig. 2.

We understand that on the journey from Newcastle to London, on Sunday, 30 July, the running time of this car was 5 hours 48 minutes, with eight stops for signal checks. The non-stop run for the « Flying Scotsman » over the same distance occupies 4 hours 55 minutes. The fuel consumption was 35 gallons. The maximum speed with the 90-H.P. engine is said to be 60 m.p.h., and 75 m.p.h. with the 150-H. P. engine.

[621 43 (.45)]

Italian petrol rail cars.

(The Railway Gazette.)

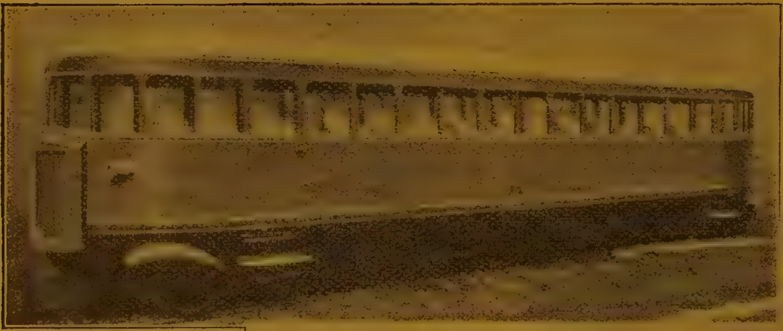


Fig. 1.
General view of car.

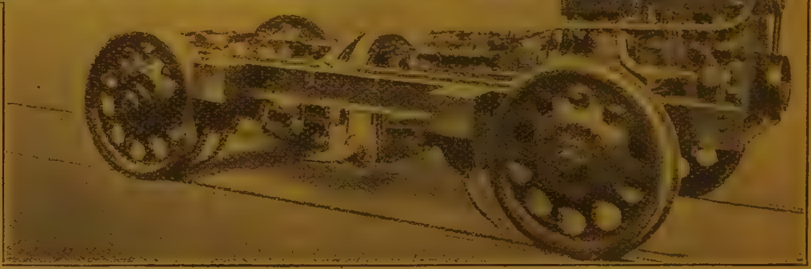


Fig. 2.
The power bogie.

The accompanying illustrations depict a double-bogie light-weight railcar, which has been tried out in both high-speed and stopping services on the Italian State Railways during recent months, and which, by reason of its successful working is to form the prototype of a number of similar vehicles just ordered by the Italian State Railways authorities. Built at the Fiat Lingotto works, the car has a six-cylinder petrol engine developing a maximum of 130 B.H.P. at 2 000 r.p.m., and with a fuel capacity of 250 litres (55 Imp. gallons), the radius of action is some 420 miles. The aluminium sides of the car are built-up on a framework of electrically-welded steel sections, and the flooring of the passenger compartment is also of aluminium, corrugated sheets with a filling of wood being employed.

Both bogies are of a patented type, the driving bogie carrying the engine and four-speed transmission *en bloc*, as shown in figure 2. The bogies have double suspension, and further to assist comfortable riding, rubber dampers are used between each bogie and the body. Roller bearings are employed for the axle, and the eight wheels are fitted with internally-expanding brakes actuated by compressed air. The various engine control handles are mounted in each driving compartment, the engine controls being mechanical, and those of the gearbox, clutch, reversing gear, and freewheel, pneumatic.

During the course of the trials the

car ran over the Turin-Saluzo, Turin-Alessandria, Turin-Florence-Rome, and Rome-Formio lines at speeds up to 75 m. p.h., the fuel consumption averaging 222 gr. (0.49 lb.) per B.H.P.-hour, and the transmission showing an efficiency up to 90 %. The car is now understood to be in service on the Turin-Aosta section, on which it covers the 42 miles between Ivrea and Aosta in 97 minutes one way and in 83 the other, including 16 booked intermediate stops in each direction. This compares with the schedule for normal steam stopping trains of 138 and 126 minutes respectively.

The characteristics of the three types of Fiat car now on order are as given in the following table :

Maximum B.H.P.	130.	130	260.
Overall length	49 ft. 6 in.	57 ft. 9 in.	72 ft. 2 in.
Spacing of bogie centres	32 ft. 4 in.	40 ft. 6 in.	54 ft. 1 in.
Bogie wheelbase	9 ft. 2 in.	9 ft. 2 in.	10 ft. 2 in.
Diameter of wheels	3 ft.	3 ft.	3 ft.
Seating accommodation	48.	64	80.
Tare weight	13.85 tons.	15.26 tons.	18.65 tons.

656. 1 (43) & 656. 261 (43)]

Transporting rail wagons by road in Germany.

(Modern Transport.)

Of recent years railway companies throughout the world have adopted various measures to reduce to a minimum the time and labour entailed in freight handling. Rapid progress has been made in this country in the use of containers and, whilst these, as a rule, are transferred by cranes at the terminal points, appliances have been introduced which enable the containers to be rolled from rail to road vehicle and *vice versa*. Realising that there are many factories without rail connection and, at the same time, constantly requiring wagon loads of traffic, the German State Railway Company has for some time been experimenting with a view to finding the most suitable means of transferring rail wagons, as distinct from containers, from goods terminal to consignees' premises.

A recent issue of *Die Reichsbahn*, the official organ of the German State Railways, contains a most interesting article by Herr Hans Von Culemeyer, a leading official of the Company, describing the latest methods which have been adopted in Germany for this important phase of through transit. By the courtesy of Herr Culemeyer we are enabled to reproduce a series of photographs depicting a 16-wheeled bogie trailer which has been specially designed for carrying 20-ton railway trucks. The vehicle in this instance was built by the Gothaer Waggonfabrik A. G., of Gotha, from designs and structural details supplied by the chief construction office of the Reichsbahn, and during trials has given most satisfactory results. As will be seen from the photographs, the trailer consists of two eight-wheeled bogies which are connected by means of an ingenious type of telescopic underpin. The vehicle is fitted with the usual railway type of compressed-air brake acting on the eight wheels of one of the bogies. It is hauled by a petrol-engined tractor equipped with a winch whereby the trailer can be drawn into position end-on to a loading platform. Thus, when it is required to load a railway

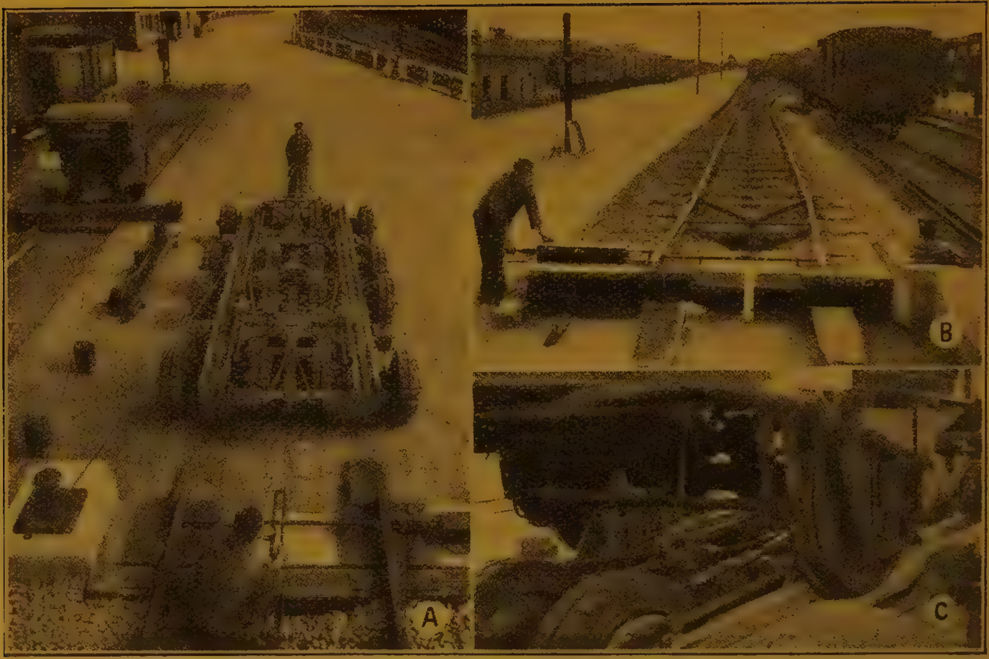


Fig. 1. — A 16-wheel double-bogie trailer is used by the German Railway Administration for conveying rail wagons through the streets : *a*) Manœuvring the trailer into position at the rail loading platform; *b*) end-on view of the platform; *c*) method of anchoring rail wagon axle to trailer.

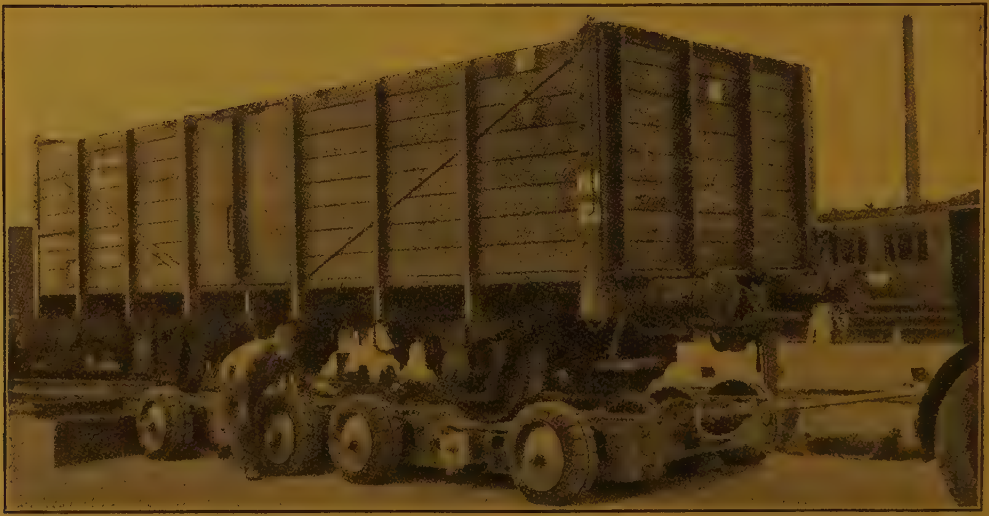


Fig. 2. — The leading wheels of the rail wagoı having been drawn on to the first bogie, the latter is hauled forward by the tractor unti the rear wheels rest on the second bogie.

truck on to the trailer the two bogies, closely coupled together, are drawn towards the ramp by a rope operating on the winch through the intermediary of a capstan in the manner shown in figure 1. The platform is 56 cm. (1 ft. 10 in.) high and the ends of the rails are hinged so as to enable them to be quickly brought into alignement with the rails on the road vehicle. As will be seen from figure 1, the loading arrangements are quite simple and are such as can be provided at little cost at goods depots and factories.

At the foot of the platform are two small ramps upon which run rollers situated below and at each end of the framework of the trailer, the latter being thereby prevented from sinking when a wagon is being drawn on to it. For loading purposes both bogies are coupled tightly together so that the rail bearers form a continuous runway for the oncoming wagon. Moreover, the mounting of the two bogies on eight axles and sixteen rubber-tyred wheels enables steering to be easily accomplished by hand from the drawbar in front or rear. Thus, when the bogies are being drawn by the tractor towards the loading platform, the rear drawbar is utilised and the whole outfit can be steered by one man, and, on reaching the platform, the end is securely fixed in position by the use of an ordinary wagon coupling, the process being shown in figure 1.

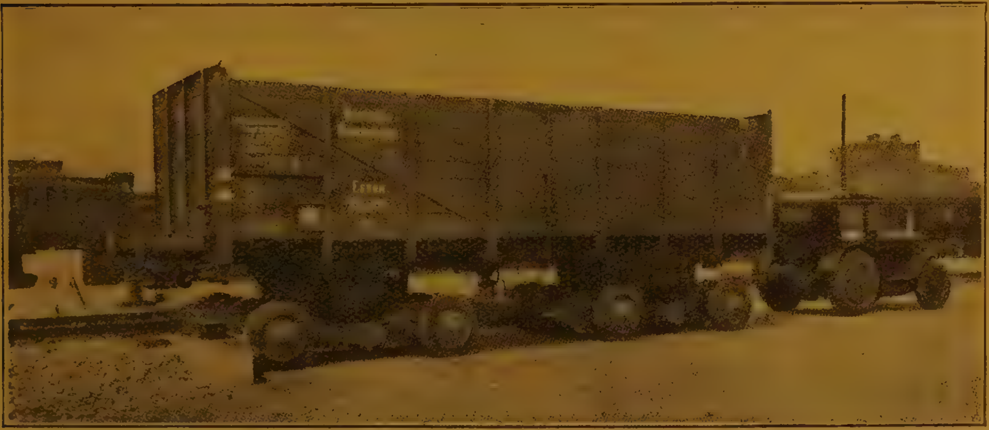


Fig. 3. — The rail wagon in position on the two bogies is now ready for haulage through the streets.

This operation being accomplished, the wagon — in this instance a 20-tonner — is drawn by the capstan and cable equipment of the tractor on to the road unit until its leading pair of wheels is resting on the middle of the first bogie (fig. 2). A lever is then moved to actuate two axle guards on the bogie frame, which are raised so that they rest beneath the axle of the wagon, thus anchoring it in position, as shown in figure 1 *c*). Next, the coupling between the two bogies is disconnected and the first bogie is drawn forward until the rear wheels of the goods wagon rest on the middle of the second bogie. During this operation the telescopic connecting pin has been automatically drawn out between the two bogies, the wheelbase being thereby increased from 3 m. (9 ft. 10 in.) to 4.50 m. (14 ft. 8 1/2 in.), a distance sufficient to embrace the majority of goods wagons used on the German Railways. For wagons of larger capacity, however, the connecting rod can be lengthened by the insertion of an additional piece to provide a maximum wheelbase of 8 m. (26 ft. 3 in.).

The rear axle of the goods wagon having been anchored in position similarly to the leading axle, the whole wagon is then lowered by 21 cm. (8 1/4 in.) to a height of 30 cm. (11 3/4 in.) above ground level, each pair of wheels being so lowered in from 5 to 10 seconds by releasing a set of valves. The brake pipes are then coupled between the two bogies and connected up with the tractor, and the tractor-trailer combination is then ready for the road.

It is pointed out that, owing to the remarkably low centre of gravity thus achieved, road transport of heavy rail wagons is rendered perfectly safe and, by reason of the large number of wheels, depressions in the road surface are of no moment, so that the journey can be made without damage to the goods. When travelling, the road vehicle is practically noiseless and, owing to the excellence of the springing and arrangement of the load, vibrations are hardly perceptible.

[621 .138.1 (.42)]

Pneumatic ash extraction plant on the London Midland & Scottish Railway.

(*Modern Transport.*)

Vacuum-cleaning a locomotive is aptly descriptive of a method of cleansing smoke-boxes which has been adopted by the London Midland and Scottish Railway at that company's locomotive depot at Willesden, N. W. This installation, the first of its kind, has been manufactured and installed by Davidson and Co., Ld., of the Sirocco Engineering Works, Belfast, Northern Ireland, who, while perhaps better known for the manufacture of « Sirocco » multi-bladed fans and allied equipment, have succeeded in applying the principles of pneumatic conveyance in a very efficient and practical manner for this purpose.

An ertswhile cogent argument of those two favour wholesale electrification of our railways was that the steam locomotive must of necessity spend a considerable portion of its life in periodical and frequent preparation for useful activities. Coaling, turning, replenishment of sand-boxes, tube sweeping, emptying smoke-boxes and ash cleansing are operations many of which require daily attention, and, while the design and performance of steam engines have, during the last decade, shown marked improvement, the daily work of maintenance has not diminished to any material degree. A noticeable step forward has been taken of recent years in the direction of locomotive coal handling installations, of which particulars have from time to time been published in *Modern Transport*, and now we are enabled to record a further development in the introduction of equipment designed for cleansing the smoke-boxes of locomotives.

General arrangement.

The « Sirocco » pneumatic ash extraction plant at Willesden, which has been installed to the requirements of Mr. W. A. Stanier, chief mechanical engineer, L. M. S. R., is designed to deal with two locomotives simultaneously. It has a total capacity for dealing with ashes and similar material at the rate of 6 tons an hour, or, roughly, 1 cwt. a minute, from

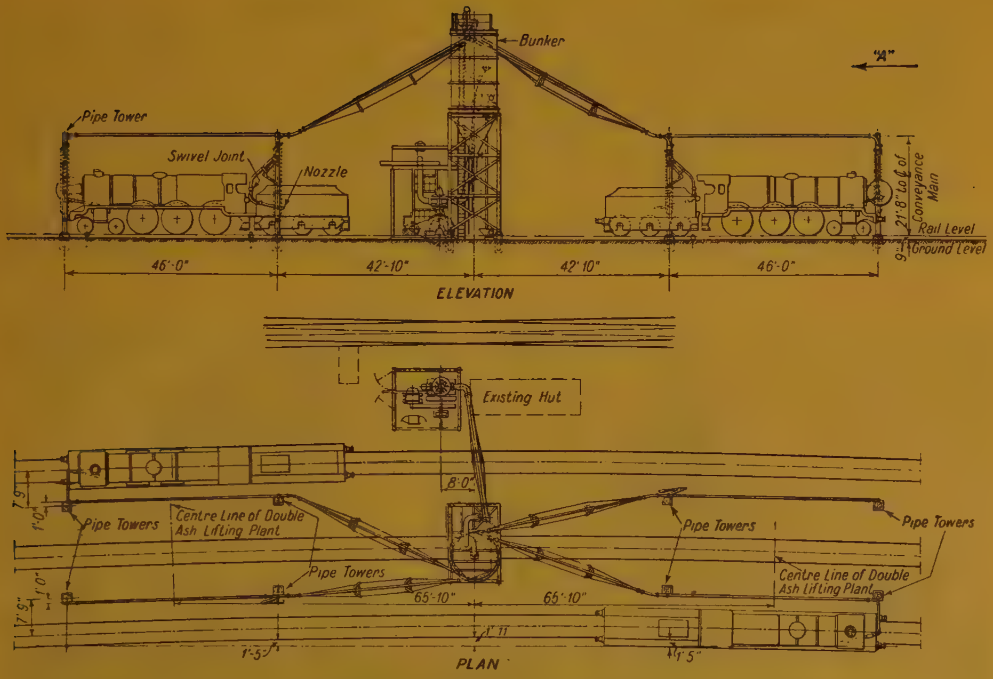
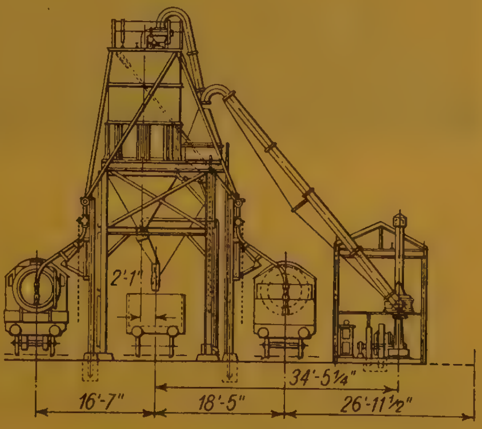


Fig. 1. — General arrangement of « Sirocco » pneumatic ash extraction plant, Willesden depot, London Midland & Scottish Railway.

each engine being cleaned. Four operating positions are arranged, two each on adjacent parallel tracks, and each position is serviced by two extraction pipes, either of which is used according to whether the engine comes in forward or reverse, details being shown in the accompanying drawing.

Suction nozzles of special non-choking design, and directed by hand as required, extract the ashes at the points to be cleansed. The material is then conveyed by means of pivoted pipes, having ball and socket joints, and supported on swivelling jibs attached to suitable frames mounted alongside the track, to an air-tight receiver, of 1 350 cubic feet (approximately 13 tons) capacity, whereby preliminary separation of the ash from the conveying air is carried out.



View looking in direction of arrow "A" in fig. 1.

Fig. 2.

Operation.

The conveying stream, after passage through the receiver, where it has been partially relieved of matter in suspension, then passes into the volute of a « Davidson » « D » type patent centrifugal dust collector, whence it emerges in almost a dust-free condition.

After passage through the centrifugal collector the air, now practically dust-free, is treated in an air washer to remove the last traces of suspended matter. The water sprays within the washer are brought into action automatically by vacuum-operated valves when the plant is set to work, the dirty water being discharged continuously by means of a barometric pipe. The ashes contained in the receiver and recovered from the dust collector, whence they pass into a separate compartment in the main receiver, are finally discharged by a vertical chute into wagons placed for this purpose beneath the appliance. A chain-controlled discharge valve and flexible hose at the base of the receiver serve to release the ashes into the wagon below when necessary.

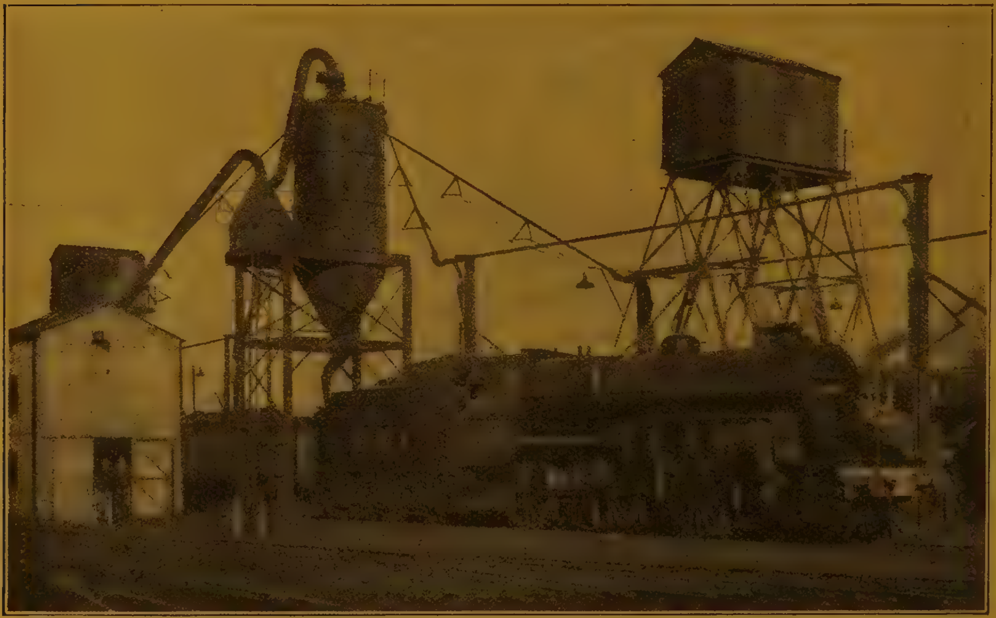


Fig. 3. — A locomotive at the ash extraction plant at Willesden.

The installation operates at a pressure of about 5 lb. per sq. inch below that of the atmosphere. This pressure is set up and maintained by a vacuum exhauster of the vertical double-acting, single-cylinder reciprocating type, connected on the outlet side of the air washer. The exhauster, driven by means of vee ropes from an electric motor, is equipped with spring-loaded valves entirely automatic in action. No direct contact between the piston and the cylinder wall occurs, so that wear of these parts is practically eliminated.

Results of working.

We are informed that, since the installation was put into regular service some months ago, no manual handling of smoke-box ash has been necessary from the time the locomotives arrive until the extracted material is carried away in wagons. Time and labour are substantially reduced, while, at the same time, the presence of clouds of floating dust, formerly a source of annoyance and irritation, is eliminated. No wastage of ashes in the vicinity of the yard occurs, and maximum recovery of this material in the dry state, when it possesses a not inconsiderable calorific value, can be made for the purpose of firing different types of stand-by plant. Furthermore, the very thorough cleansing which has been made possible by the use of the plant assists in the detection of latent defects. There is undoubtedly considerable scope for the use of this type of labour-aiding appliance at locomotive depots, and the L. M. S. is to be commended on being the first to test its merits.

[313. 625. 143. 3]

Statistics of rail breakages for the year 1932.

We publish hereafter, in the form adopted at the Madrid Congress (1930) (1), the information supplied by member Administrations in connection with the rail fractures which occurred on their lines in 1932.

In the tables hereafter, and unless stated otherwise (2):

Light rails *applies to rails of a weight less than 85 lb. per yard (42.5 kgr. per metre),*

Medium rails, *to rails of 85 to 105 lb. per yard (42.5 to 52.5 kgr. per metre),*

Heavy rails, *to those weighing 106 lb. per yard (53 kgr. per metre) or over.*

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(1) See *Bulletin of the Railway Congress*, December 1930, p. 2236, 2240-2242.

(2) See *Bulletin of the Railway Congress*, March 1926, p. 240.

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	AGE OF RAILS : Less than 5 years.			5 to 10 years.			10 to 15 years.			15 to 20 years.			More than 20 years.			The whole of the rails.			*Maximum axle load.*
	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
ARGENTINE. Buenos Ayres and Pacific Railway *		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
A. Rails outside tunnels *Light* . .	...	...	...	...	...	...	...	...	...	...	...	...	3	222.5	8.42	3	222 5	8.42	13.8
A. Rails outside tunnels *Medium* .	...	...	...	...	...	...	...	...	...	...	...	...	16	997.6	10.02	16	997.6	10.02	17.8-18.8
A. Rails outside tunnels *Heavy*.	...	...	...	2	1 820.2	0.69	...	...	...	...	...	...	6	1 820.2	2.06	8	1 820.2	2.74	18.8-21.3
Total . . .	...	...	...	2	1 820.2	0.69	...	...	...	...	...	...	25	3 040 3	5.13	27	3 040.3	5.55	

	Percentage of fractures in the part: covered by the fishplates	clear of the fishplates	NUMBER OF FRACTURES : on straight lines or curves of > 800 m. (40 chains) radius	on curves of ≤ 800 m. (40 chains) radius: Lower rail.	Higher rail.	on a rising or falling gradient: ≤ 10 mm. per m. (1 in 100)	> 10 mm. per m. (1 in 100)
			← Level track →				
D. *Light rails.* . .	2 %	1 %	1	...	...	2	...
D. *Medium rails* . .	4 %	12 %	3	...	...	13	...
D. *Heavy rails* .	1 %	7 %	4	...	...	4	...
		Total . . .	8	...	...	19	...

E.		*Light.*	*Medium*	*Heavy.*
a) New clean fractures	with internal transverse fissure	1	...	...
	without internal transverse fissure	1	8	5
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head	in the head	1	9	4
	in the foot	...	2	1
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head . . .	in the web	1	1	1
d) Number of pieces rails are broken into	. . . .	2+3	2+3	2+3

Note. — *a*) Light rails = from 50.4 to 54.7 lb. per yard.
Medium rails = from 70.1 to 85 lb. per yard.
Heavy rails = 100 lb. per yard.

b) No tunnels on this System.

* Running lines only.

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	Length of single track with each rail profile.	Age of rails :										
		5 years and less.						6 to 10 years.				
		Year of manufacture.	Number of fractures			Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Year of manufacture.	Number of fractures			Length of single track of this class.
			In the length of the rail.	At the joint.	Total.				In the length of the rail.	At the joint.	Total.	
1	2	3	4	5	6	7	8	9	10	11	12	13
	Miles.					Miles.						Miles.
BELGIUM.												
Belgian National Railways Company. **(Year 1931).**												
Light rails :												
Weight : 38 kgr. (76.60 lb. per yard), profile adopted in 1863 (steel) and given up since.	82.7	...	...	...	...	...	...	...	...	...	...	...
Weight : 40.65 kgr. (81.94 lb. per yard), profile adopted in 1898 and given up since.	1 113.5	...	...	...	...	...	...	1922 to 1926	...	2 (*1)	2	301.0
Medium rails :												
Weight : 50 kgr. (100.79 lb. per yard), profile adopted in 1910.	2 848.4	1927 to 1931	...	9	9	1 153.4	4.85	1922 to 1926	10	50 (*6)	60	1 130.8
Weight : 52 kgr. (104.82 lb. per yard), profile adopted in 1886 and given up since.	110.4	...	...	...	...	...	...	...	...	...	...	...
Heavy rails :												
Weight : 57 kgr. (114.90 lb. per yard), profile adopted in 1907 and given up since	176.5	...	...	...	...	...	...	...	...	...	...	...
Total. . . .	4 331.5 including 36.9 in tunnels.		...	9	9	...	...	...	10	52 (*7)	62	...

Number of English tons gross) hauled, passengers and goods : 16 985 153 800.

Number of train-miles : 43 312 428.

Total number of fractures : 254.

Number of fractures per 10 000 000 train-kilometres or 6 250 000 train-miles : 36.4.

(*) In tunnels.

	Age of rails :															Total number of fractures for the whole of the System.	*Maximum axle load.*
	10 to 15 years.					15 to 20 years.				over 20 years.							
	Number of fractures.			Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Year of manufacture.	Number of fractures.			Year of manufacture.	Number of fractures.			Length of single track with profiles 15 years old and over.	Number of fractures per 1 000 km. or per 625 m. of these classes (rails 15 years old and over).		
e.	In the length of the rail.	At the joint.	Total.				In the length of the rail.	At the joint.	Total.		In the length of the rail.	At the joint.	Total.				
	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32
				Miles.										Miles.			Pounds.
	...	...	...	...	...	...	...	...	...	1864 to 1898	4	...	4	82.7	30.0	4	52.250
921	...	1	1	22.9	27.0	...	...	...	...	1898 to 1911	17	25 (*1)	42	789.6	33.0	45 (*2)	
921	4	19	23	253.4	56.37	1912 to 1914	2	33	35	1910 to 1911	4	25 (*2)	29	310.8	128.0	156 (*8)	
	...	...	...	...	...	...	...	...	...	1887 to 1908	4	4	8	110.4	44.90	8	
	...	...	...	...	...	1912 to 1914	1	14	15	1907 to 1911	4 (*2)	34 (*4)	26	176.5	144.0	41 (*6)	
	4	20	24	...	...	...	3	47	50	...	33 (*2)	76 (*7)	109	1 470.0	...	254 (*16)	

Percentages of fractures		At the joint.	Outside the joint.
	I. *Light rails*	57.2 %	42.8 %
	II. *Medium rails*	85.4 %	14.6 %
	III. *Heavy rails*	87.6 %	12.4 %

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	Length of single track with each rail profile.	Age of rails :										
		5 years and less.						6 to 10 years.				
		Year of manufacture.	Number of fractures			Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Year of manufacture.	Number of fractures			Length of single track of this class.
			In the length of the rail.	At the joint.	Total				In the length of the rail.	At the joint.	Total.	
1	2	3	4	5	6	7	8	9	10	11	12	13
	Miles.					Miles.						Miles.
BELGIUM.												
Belgian National Railway Company. **(Year 1932)**												
Light rails :												
Weight : 38 kgr. (76.60 lb. per yard), profile adopted in 1863 (steel) and given up since.	78.8	...	...	...	...	...	...	...	...	...	...	...
Weight : 40.65 kgr. (81.94 lb. per yard), profile adopted in 1898 and given up since.	1 046.1	1925	...	...	...	...	...	1923 to 1927	...	...	...	239.8
Medium rails :												
Weight : 50 kgr. (100.79 lb. per yard), profile adopted in 1910.	2 970.4	1928 to 1932	2	8 (*2)	10	1 185.4	5.24	1923 to 1927	3	78 (*8)	81	1 177.0
Weight : 52 kgr. (104.82 lb. per yard), profile adopted in 1886 and given up since.	78.4	...	...	...	...	...	...	...	...	...	...	...
Heavy rails												
Weight : 57 kgr. (114.90 lb. per yard), profile adpoted in 1907 and given up since.	151.0	...	...	...	...	...	...	...	...	...	...	...
Total. . .	4 324.7 including 36.9 in tunnels		2	8 (*2)	10	...	...	...	3	78 (*8)	81	...

Number of English tons (gross) hauled, passengers and goods : 14 176 286 000.

Number of train-miles : 40 645 500.

Total number of fractures : 219.

Number of fractures per 10 000 000 train-kilometres or 6 250 000 train-miles : 33.47.

(*) In tunnels.

Age of rails :																	Total number of fractures for the whole of the System.	*Maximum axle load.*
10 to 15 years.						15 to 20 years.				Over 20 years.								
ear o anu- cture.	Number of fractures.			Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Year of manu- facture.	Number of fractures.			Year of manu- facture.	Number of fractures.			Length of single track with profiles 15 years old and over.	Number of fractures per 1 000 km. or per 625 m. of these classes (rails 15 years old and over)			
	In the length of the rail.	At the joint.	Total.				In the length of the rail.	At the joint.	Total.		In the length of the rail.	At the joint.	Total.					
15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30		31	32
				Miles.										Miles.				Pounds.
...	...	...	...	...	...	...	...	...	...	1864 to 1898	1	...	1	78 8	7 87		1	52 250
9 to 1922	...	1	1	83.9	7.40	...	...	...	...	1898 to 1912	9	12	21	722.4	18.9		22	
9 to 1922	4	28	32	409.9	48.48	1913 and 1914	...	11	11	1910 to 1912	...	30	30	198.1	128.00		164 (*10)	
...	...	...	...	...	...	...	...	...	...	1886 to 1908	2	8	10	78.4	79.0		10	
...	...	...	...	...	...	1913 and 1914	...	...	...	1907 to 1910	...	22 (*3)	22	151.0	90.53		22 (*3)	
...	4	29	33	...	...	...	...	11	11	...	12	72 (*3)	84	...	...		219 (*13)	

		At the joint.	Outside the joint.
Percentages of fractures	I. *Light rails*	59 %	41 %
	II. *Medium rails*	93.6 %	5.4 %
	III. *Heavy rails*	100 %	0 %

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	AGE OF RAILS : Less than 5 years. Number of fractures.	Less than 5 years. Length of single track of this class.	Less than 5 years. Number of fractures per 1 000 km. or per 625 miles.	5 to 10 years. Number of fractures.	5 to 10 years. Length of single track of this class.	5 to 10 years. Number of fractures per 1 000 km. or per 625 miles.	10 to 15 years. Number of fractures.	10 to 15 years. Length of single track of this class.	10 to 15 years. Number of fractures per 1 000 km. or per 625 miles.	15 to 20 years. Number of fractures.	15 to 20 years. Length of single track of this class.	15 to 20 years. Number of fractures per 1 000 km. or per 625 miles.	More than 20 years. Number of fractures.	More than 20 years. Length of single track of this class.	More than 20 years. Number of fractures per 1 000 km. or per 625 miles.	The whole of the rails. Number of fractures.	The whole of the rails. Length of single track of this class.	The whole of the rails. Number of fractures per 1 000 km. or per 625 miles.	*Maximum axle load.*
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
BELGIUM. CONGO COLONY. Lower Congo to Katanga Railway.		Miles.			Miles.			Miles.			Miles.						Miles.		English tons.
A Rails outside tunnels. **Bukama-Sakania line.** 29.2 kgr. (59.2 lb.)	1	93.2	6	1	24.9	25	2	201.9	6	7	248.5	18	...	...	...	11	568.5	12	14.8
Tenke-Dilolo line. (L. K. D.) 29.3 kgr. (59.2 lb.)	1	338.6	2	...	...	...	...	...	...	...	...	...	...	...	...	1	338.6	2	14.8

Port Francqui-Bukama line (L. K. D.) : No fractures.

Number of train-miles { Bukama-Sakania : 447 400.
Tenke-Dilolo : 174 000.
Port Francqui-Bukama : 303 230.

Number of Engl. ton-miles { Bukama-Sakania : 129 650 000 gross — 59 900 000 nett.
Tenke-Dilolo : 29 355 260 gross — 12 231 360 nett.
Port Francqui-Bukama : 67 272 500 gross — 26 909 000 nett.

	Percentage of fractures in the part: covered by the fishplates	Percentage of fractures in the part: clear of the fishplates	NUMBER OF FRACTURES : on straight lines or curves of > 800 m. (40 chains) radius	on curves of ≤ 800 m. (40 chains) radius: Lower rail.	on curves of ≤ 800 m. (40 chains) radius: Higher rail.	on a rising or falling gradient: ≤ 10 mm. per m. (1 in 100)	on a rising or falling gradient: > 10 mm. per m. (1 in 100)
D. *Light rails* I*	27.2	72.8	4	2	5	3	8
D. *Light rails* II**	100	...	1	...	...	1	...

Miles of single track { Bukama-Sakania : 568.5 miles . . . .
Tenke-Dilolo : 338.6 miles. . . . .
Port Francqui-Bukama : 739.4 miles . . . . . } Total : 1 646.5 miles.

E.	*Light rails.* I.	*Light rails.* II.
a) New clean fractures { with internal transverse fissure	...	...
a) New clean fractures { without internal transverse fissure	1	...
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head . . . . } in the foot	1	...
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head } in the head	7	1
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head . . . } in the web	2	...
d) Number of pieces rails are broken into	2 pieces : 4 Cracked : 7	Cracked.

Number of fractures :

Total { I : 11. II : 36.

Per 10 000 000 tr.-km. or 6 250 000 train-miles { I : 153. II : 36.

Per 1 billion tkm. (gross) or 612 000 000 Engl. ton-miles (gross) { I : 52. II : 21.

Per 1 billion tkm. (nett) or 612 000 000 Engl. ton-miles (nett) { I : 112. II : 50.

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	Less than 5 years.			5 to 10 years.			10 to 15 years.			15 to 20 years.			More than 20 years.			of the rails.			*Maximum axle load.*
	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
DENMARK.		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		
State Railways.																			
A. Rails outside tunnels *Light*	...	93.8	...	1	85.6	7.3	1	164.4	3 8	3	187.8	9.9	118	640.3	114	123	1 171.9	65.0	...
Medium	...	178.8	...	2	75.7	16.4	4	174.8	14.2	1	51.1	12.1	7	226.6	19.2	14	707.0	12.3	...
Total	...	272.6	...	3	161.3	11.6	5	339.2	9.2	4	238.9	10.4	125	866.9	90	137	1 878.9	45	...
B. Rails in tunnels *Light*	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
Medium	...	...	...	...	...	...	...	...	...	2	3.7	333	...	...	...	2	3.7	333	...
Total	...	...	...	...	...	...	...	...	...	2	3.7	333	...	...	...	2	3.7	333	...
C. The whole of A and B *Light*	...	93.8	...	1	85.6	7.3	1	164.4	3.8	3	187.8	9.9	118	640.3	114	123	1 171.9	65.0	...
Medium	...	178.8	...	2	75.7	16.4	4	174.8	14.2	3	54.8	34	7	226.6	19.2	16	710.7	15.3	...
Total	...	272.6	...	3	161.3	11.6	5	339.2	9.2	6	242.6	15.4	125	866.9	90	139	1 882.6	45.8	...

Number of train-miles: 15 307 630.

Number of English ton-miles: 2 770 545 000.

Number of fractures { total: 139.
per 10 000 000 tr.-km. or 6 250 000 train-miles: 57.
per 1 billion tkm. or 612 000 000 English ton-miles: 31.

	Percentage of fractures in the part		NUMBER OF FRACTURES:				
			on straight lines or curves of > 800 m. (40 chains) radius	on curves of ⩽ 800 m. (40 chains) radius		on a rising or falling gradient	
	covered by the fishplates	clear of the fishplates		Lower rail.	Higher rail.	⩽ 10 mm. per m. (1 in 100)	> 10 mm. per m. (1 in 100)
D. *Light rails.*	71 %	29 %	...	...	...	...	...
Medium rails.	75 %	25 %	...	...	...	...	...
		Total	...	...	...	...	...

		Light rails.	*Medium rails.*
E. *a*) New clean fractures	with internal transverse fissure	9	0
	without internal transverse fissure	18	4
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head	in the foot	21	7
	in the head	16	1
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head	in the web	59	4
d) Number of pieces rails are broken into	2 pieces	109	14
	3 —	12	
	> 3 —	2	20

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	AGE OF RAILS :															The whole of the rails.			*Maximum axle load.*
	Less than 5 years.			5 to 10 years.			10 to 15 years.			15 to 20 years.			More than 20 years.						
	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
FRANCE. State Railways. (*)		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
A. Rails outside tunnels. *Light.*	...	5.0	...	...	44.7	...	...	40.4	...	...	185.8	...	200	4 232.2	29	200	4 508.1	28	
Medium.	9	1 075.0	5	14	840.1	10	2	333.1	4	6	843.2	4	17	433.7	24	48	3 525.1	8	
Heavy	...	12.4	...	...	3.1	...	...	...	...	...	...	...	...	...	...	...	15.5	...	
Total	9	1 092.4	5	14	887 9	10	2	373.5	3	6	1 029.1	4	217	4 6 5.9	29	248	8 048.7	19	
B. Rails in tunnels. *Light.*	...	...	...	...	...	...	...	0.6	...	...	...	...	1	1.9	333	1	2.5	250	
Medium.	...	0.6	...	...	1.9	...	...	3.7	...	...	4.4	...	...	0.6	...	...	11.2	...	19.7
Heavy	...	23 6	...	7	14.3	304	...	2.5	...	...	...	...	...	...	...	7	40.4	108	
Total	...	24.2	...	7	16.2	269	...	6.8	...	...	4.4	...	1	2.5	250	8	54.1	92	
C. The whole of A and B. *Light.*	...	5.0	...	...	44.7	...	...	41.0	...	...	185.8	...	201	4 234.1	29	201	4 510.6	28	
Medium.	9	1 075.6	5	14	842.0	10	2	336.8	4	6	847.6	4	17	434.3	24	48	3 536.3	8	
Heavy	...	36.0	...	7	17.4	250	...	2.5	...	..	...	...	...	...	...	7	55.9	78	
Total	9	1 116.6	5	21	904.1	14	2	380.3	3	6	1 033.4	4	218	4 668.4	29	256	8 102.8	20	

Number of train-miles : 43 612 362.
Number of English ton-miles : 14 259 714 500.

Number of fractures { total : 256.
per 10 000 000 tr.-km. or 6 250 000 train-miles : 36.
per 1 billion tkm. or 612 000 000 English ton-miles : 11.

	Percentage of fractures in the part		NUMBER OF FRACTURES :				
	covered by the fishplates	clear of the fishplates	on straight lines or curves of > 800 m. (40 chains) radius	on curves of ⩽ 800 m. (40 chains) radius		on a rising or falling gradient	
				Lower rail.	Higher rail.	⩽ 10 mm. per m. (1 in 100)	> 10 mm. per m. (1 in 100)
D. *Light rails.*	26 %	74 %	131	44	26	174	27
Medium rails.	58 %	42 %	46	1	1	46	2
Heavy rails	100 %	...	7	...	...	6	1
Total			184	45	27	226	30
Miles of single track of each class.			6 945.2	1 157.6		7 067.6	1 035.2

		Percentage		
		Light	*Medium*	*Heavy*
E. *a)* New clean fractures	with internal transverse fissure	1	2	...
	without internal transverse fissure	45	25	...
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head	in the foot	21	35	...
	in the head	14	18	57
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head	in the web	[illegible]	[illegible]	[illegible]

d) Number of pieces rails are broken into	En rails :		
	Light	*Medium*	*Heavy*
2 pieces	175	41	6
3 —	20	6	1
4 —	4	1	»
5 —	»	»	»
6 —	»	»	»
7 —	»	»	»
8 —	»	»	»

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	Less than 5 years.			5 to 10 years.			10 to 15 years.			15 to 20 years.			More than 20 years.			of the rails.			*Maximum axle load.*
	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Alsace-Lorraine Railways, including the Guillaume-Luxembourg Lines		Miles			Miles			Miles.			Miles.			Miles			Miles.		English tons.
A. Rails outside tunnels. *Light.*	...	...	...	...	16.8	...	4	12.4	200	9	171.8	33	53	1128.9	29	66	1329.9	31	Locos : 18.4 — Wagons : 15.8
Medium.	5	352.2	9	18	287.7	39	6	79.8	47	5	65.2	48	10	225.6	28	44	1010.5	27	
Heavy .	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	
Total . . .	5	352.2	9	18	304.5	37	10	92.2	67	14	237.0	37	63	1354.5	29	110	2340.4	29	
B. Rails in tunnels. *Light.* .	...	...	...	...	...	...	...	0.6	...	1	4.0	153	4	3.1	800	5	7.7	400	
Medium.	1	6.8	91	3	2.5	750	...	0.3	...	...	1.9	...	1	0.6	1 000	5	12.1	256	
Heavy .	...	9.6	...	...	...	...	...	...	...	...	...	...	...	...	...	...	9.6	..	
Total . . .	1	16.4	38	3	2.5	750	...	0.9	...	1	5.9	105	5	3.7	833	10	29.4	211	
C The whole of A and B. *Light.* .	...	...	...	...	16.8	...	4	13.0	190	10	175.8	35	57	1132.0	31	71	1337.6	33	
Medium.	6	359.0	10	21	290.2	45	6	80.1	47	5	67.1	46	11	226.2	30	49	1022.6	30	
Heavy .	...	9.6	...	...	...	...	...	...	...	...	...	...	...	...	...	...	9.6	...	
Total . . .	6	368.6	10	21	307.0	43	10	93.1	67	15	242.9	38	68	1158.2	31	120	2369.8	31	

Number of train-miles : 17 214 375.
Number of English ton-miles : 7 070 859 300.

Number of fractures } per 10 000 000 tr.-km. or 6 250 000 trains-miles : 43.
per 1 billion tkm. or 612 000 000 English ton-miles : 10.

Note. — This table does not include fractures of *machined* rails in points and crossings, i. e. those of the nose, wing rails, point blades and stock rails.

	Percentage of fractures in the part		NUMBER OF FRACTURES :				
			on straight lines or curves of > 800 m. (40 chains) radius	on curves of ⩽ 800 m. (40 chains) radius		on a rising or falling gradient	
	covered by the fishplates	clear of the fishplates		Lower rail.	Higher rail.	⩽ 10 mm. per m. (1 in 100)	> 10 mm. per m. (1 in 100)
D. *Light rails* . . .	71.83 %	28.17 %	40	21	10	40	5
Medium rails . .	53.06 %	46.4 %	46	2	1	38	...
Heavy rails . .	...	...	...	...	...	...	...
Total . . .			86	23	11	78	5
Miles of single track of each class.			1882.8	487.0		1633.0	208.2

	Light rails.	*Medium rails.*	*Heavy rails.*
E. *a*) New clean fractures { with internal transverse fissure	4.22 %	2.04 %	...
without internal transverse fissure	28.17 %	30.61 %	...
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head . . . . { in the foot	16.90 %	20.41 %	...
in the head	9.86 %	12.25 %	...
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head . . . { in the web	40.85 %	34.69 %	...

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	AGE OF RAILS : Less than 5 years.			5 to 10 years.			10 to 15 years.			15 to 20 years.			More than 20 years.			The whole of the rails.			*Maximum axle load.*
	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Paris Circle Railways (*)		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
A. Rails outside tunnels. *Light.*	...	0.41	...	...	0.06	...	...	0.53	...	...	0.86	...	1	20.60	30.13	1	22.46	27.63	18.9
Medium.	...	56.02	...	...	48.10	...	...	17.44	...	4	6.15	404.36	4	4.96	50.56	8	132.67	37.47	
Heavy	...	0.03	...	...	0.3	...	...	...	...	...	...	...	...	...	...	...	0.06	...	
Total . . .	...	56.46	...	...	48.19	...	...	17.97	...	4	7.01	354.92	5	25.56	121.43	9	155.19	36.03	
B. Rails in tunnels. *Light.*	...	...	...	...	...	...	...	...	...	...	...	...	...	0.25	...	...	0.25	...	
Medium.	...	0.16	...	...	0.02	...	...	0.82	...	...	...	...	...	...	...	...	1.00	...	
Heavy	...	4.40	...	...	0.88	...	...	0.47	...	...	...	...	...	...	...	...	5.75	...	
Total . . .	...	4.56	...	...	0.90	...	...	1.29	...	...	...	...	...	0.25	...	..	7.00	...	
C The whole of A and B. *Light.*	...	0.41	...	...	0.06	...	...	0.53	...	...	0.86	...	1	20.85	29.77	1	22.71	27.33	
Medium.	...	56.18	...	...	48.12	...	...	18.26	...	4	6.15	404.36	4	4.96	500.56	8	133.67	37.19	
Heavy	...	4.43	...	...	0.91	...	...	0.47	...	...	...	...	...	...	...	...	5.81	...	
Total . . .	...	61.02	...	...	49.09	...	...	19.26	...	4	7.01	354.92	5	25.81	120.24	9	162.19	34.47	

Number of train-miles : 1 742 705.
Number of English ton-miles : 751 633 150.

Number of fractures { total : 9.
per 10 000 000 tr.-km. or 6 250 000 train-miles : 32.09
per 1 billion tkm. or 612 000 000 English ton-miles : 7.52

	Percentage of fractures in the part		NUMBER OF FRACTURES :				
	covered by the fishplates	clear of the fishplates	on straight lines or curves of > 800 m. (40 chains) radius	on curves of ⩽ 800 m. (40 chains) radius		on a rising or falling gradient	
				Lower rail.	Higher rail.	⩽ 10 mm. per m. (1 in 100)	> 10 mm. per m. (1 in 100)
D. *Light rails* . . .	100 %	...	...	...	1	1	...
Medium rails . .	100 %	...	3	5	...	7	...
Heavy rails . .	...	...	...	...	...	...	...
		Total . . .	3	5	1	8	...
		Miles of single track of each class.	119.19	43.0		115.89	...

	Light rails .	*Medium rails*	*Heavy rails*
E. *a*) New clean fractures { with internal transverse fissure	...	...	...
without internal transverse fissure	...	25 %	...
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head { in the foot	100 %	...	...
in the head	...	12.50 %	...
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head { in the web	...	62.50 %	...
d) Number of pieces rails are broken into { 2 pieces	1	8	...
3 pieces			

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	AGE OF RAILS : Less than 5 years. Number of fractures.	Less than 5 years. Length of single track of this class.	Less than 5 years. Number of fractures per 1 000 km. or per 625 miles.	5 to 10 years. Number of fractures.	5 to 10 years. Length of single track of this class.	5 to 10 years. Number of fractures per 1 000 km. or per 625 miles.	10 to 15 years. Number of fractures.	10 to 15 years. Length of single track of this class.	10 to 15 years. Number of fractures per 1 000 km. or per 625 miles.	15 to 20 years. Number of fractures.	15 to 20 years. Length of single track of this class.	15 to 20 years. Number of fractures per 1 000 km. or per 625 miles.	More than 20 years. Number of fractures.	More than 20 years. Length of single track of this class.	More than 20 years. Number of fractures per 1 000 km. or per 625 miles.	The whole of the rails. Number of fractures.	The whole of the rails. Length of single track of this class.	The whole of the rails. Number of fractures per 1 000 km. or per 625 miles.	*Maximum axle load.*
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Est Railways (*).		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		
A. Rails outside tunnels. *Light.*	...	49.1	...	...	52.8	...	...	131.0	...	...	51.6	...	130	2342.1	34.4	130	2626.6	30.7	
Medium.	10	854.1	7.2	1	528.5	1.2	4	391.6	6.3	1	309.5	2	100	1241.2	50.05	116	3324.9	21.6	
Heavy	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	
Total . . .	10	903.2	6.9	1	581.3	1.06	4	522.6	4.7	1	361.1	1.7	230	3583.3	39.9	246	5951.5	25.7	
B. Rails in tunnels. *Light.*	...	...	...	...	...	...	...	0.1	...	...	...	...	...	4.4	...	...	4.5	...	...
Medium.	...	1.0	...	...	1.5	...	1	2.4	263.1	...	...	...	...	3.9	...	1	8.8	71.4	
Heavy	...	9.3	...	1	16.2	38.4	1	1.7	370.3	...	...	...	...	...	...	2	27.2	45.7	
Total . . .	...	10.3	...	1	17.7	35.2	2	4.2	299.8	...	...	...	...	8.3	...	3	40.5	46.1	
C. The whole of A and B. *Light.*	...	49.1	...	...	52.8	...	...	131.1	...	...	51.6	...	130	2346.5	34.4	130	2631.1	30.7	
Medium.	10	855.1	7.2	1	530.0	1.2	5	394.0	7.8	1	309.5	2	100	1245.1	49.9	117	3333.7	21.8	
Heavy	...	9.3	...	1	16.2	38.4	1	1.7	370.3	...	...	...	...	...	...	2	27.2	45.7	
Total . . .	10	913.5	6.8	2	599.0	2.07	6	526.8	7.7	1	361.1	1.7	230	3591.6	39.8	249	5992.0	25.8	

Number of train-miles : 35 781 000.
Number of English ton-miles : 15 674 872 500.

Number of fractures { total : 249.
per 10 000 000 tr.-km. or 6 250 000 train-miles : 43.2.
per 1 billion tkm. or 612 000 000 English ton-miles : 9.7.

	Percentage of fractures in the part: covered by the fishplates	Percentage of fractures in the part: clear of the fishplates	NUMBER OF FRACTURES : on straight lines or curves of > 800 m. (40 chains) radius	NUMBER OF FRACTURES : on curves of ≤ 800 m. (40 chains) radius: Lower rail.	NUMBER OF FRACTURES : on curves of ≤ 800 m. (40 chains) radius: Higher rail.	NUMBER OF FRACTURES : on a rising or falling gradient: ≤ 10 mm. per m. (1 in 100)	NUMBER OF FRACTURES : on a rising or falling gradient: > 10 mm. per m. (1 in 100)
D. *Light rails*	13	87	75	27	28	70	7
Medium rails	78	22	103	11	3	54	2
Heavy rails	100	...	2	...	...	1	...
Total . . .			180	38	31	125	9
Miles of single track of each class.			4 784 6	1 207.4		3 884.9	358.5

		Light	*Medium*	*Heavy*
		Percentage.		
a) New clean fractures	with internal transverse fissure	19.2	10.3	...
	without internal transverse fissure	46.9	24.8	...
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head	in the foot	14.6	20.5	...
	in the head	13.1	11.1	...
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head . . .	in the web	6.2	33.3	...

		Light	*Medium*	*Heavy*
d) Number of pieces rails are broken into.	2 pieces	123	93	1
	3 —	4	23	...
	4 —	3	1	1

(*) In running tracks, excluding points and crossings.

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	AGE OF RAILS : Less than 5 years.			5 to 10 years.			10 to 15 years.			15 to 20 years.			More than 20 years.			The whole of the rails.			Maximum axle load.
	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Midi Railways (*)		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
A. Rails outside tunnels. *Light.*	...	...	...	...	4.4	...	...	...	...	7	157.2	28	76	1 834.9	26	83	1 996.5	26	17.8
Medium.	4	421.3	6	18	392.7	28	46	456.1	63	2	106.9	12	...	...	...	70	1 377.0	21	
Total . . .	4	421 3	6	18	397.1	28	46	456.1	63	9	264.1	21	76	1 834.9	26	153	3 373.5	28	
B. Rails in tunnels. *Light.*	...	...	...	...	...	...	...	...	...	...	1.2	...	7	26.1	167	7	27.3	159	
Medium	...	21.1	...	7	9.3	467	1	5.0	125	2	2.4	500	...	...	..	10	37.8	164	
Total . . .	...	21.1	...	7	9.3	467	1	5.0	125	2	3.6	333	7	26.1	167	17	65.1	162	
C. The whole of A and B. *Light.*	...	...	...	...	4.4	...	...	...	...	7	158.4	20	83	1 861.0	28	90	2 023.8	28	
Medium.	4	442.4	6	25	402.0	39	47	461.1	63	4	109.3	23	...	...	...	80	1 414.8	35	
Total . . .	4	442.4	6	25	406.4	38	47	461.1	63	11	267.7	21	83	1 861.0	28	170	3 438.6	31	

Number of { train-miles : 20 291 890.
English ton-miles : 9 339 930 000.

Number of fractures { per 10 000 000 tr.-km. or 6 250 000 train-miles : 52.
per 1 billion tkm. or 612 000 000 English ton-miles : 11.

	Percentage of fractures in the part: covered by the fishplates	clear of the fishplates	NUMBER OF FRACTURES : on straight lines or curves of > 800 m. (40 chains) radius	on curves of ⩽ 800 m. (40 chains) radius: Lower rail.	Higher rail.	on a rising or falling gradient: ⩽ 10 mm. per m. (1 in 100)	> 10 mm. per m. (1 in 100)
D. *Light rails* . .	71 %	29 %	56	21	13	47	20
Medium rails . .	81 %	19 %	74	4	2	49	4
		Total . .	130	25	15	96	24
		Miles of single track of each class.	2 670.6	768.0		1 336.0	682.9

		Percentage of fractures. Light rails.	Medium rails.	Heavy rails.
E. *a*) New clean fractures	with internal transverse fissure	11 %	6 %	...
	without internal transverse fissure	33 %	36 %	...
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head	in the foot	12 %	11 %	...
	in the head	9 %	19 %	...
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head . .	in the web	35 %	28 %	...

d) Number of pieces rails are broken into	Light rails.	Medium rails.
2 pieces	74	64
3 »	9	13
4 »	6	3
5 »	1	...
6 »	...	...
7 »	...	...

* Running track, excluding joints and crossings.

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	Less than 5 years.			5 to 10 years.			10 to 15 years.			15 to 20 years.			More than 20 years.			of the rails.			*Maximum axle load.*
	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Nord Railway (*)		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
A. Rails outside tunnels. *Light.*	...	...	...	...	...	...	...	311.5	...	1	27.7	22.7	118	659.0	111.3	119	998.2	74	18 6
Medium.	9	488.2	11 4	25	610.0	25.4	30	432.6	43	1	410.1	1.5	39	1 010.7	24	104	2 951.6	21.9	
Heavy .	...	...	...	...	1.1	...	...	1.3	...	...	0.05	...	...	...	...	...	2.45	...	
Total . . .	9	488.2	11.4	25	611.1	25.4	30	745.4	25	2	437.85	2 8	157	1 669.7	58.4	223	3 952.25	35	
B. Rails in tunnels. *Light.*	...	...	...	...	...	...	...	0.1	...	...	...	...	...	...	...	...	0.1	...	
Medium.	...	0.9	...	...	0.6	...	...	3.3	...	...	0.7	...	...	1.7	...	...	7.2	...	
Heavy .	...	2.8	...	...	2.7	...	...	...	...	...	...	...	...	...	...	...	5.5	...	
Total . . .	...	3.7	...	...	3.3	...	...	3.4	...	...	0.7	...	...	1.7	...	...	12.8	...	
C. The whole of A and B. *Light.*	...	...	...	...	...	...	...	311.6	...	1	27.7	22.7	118	659.0	111.3	119	998.3	74	
Medium.	9	489.1	11.4	25	610.6	25.4	30	435.9	42.8	1	410.8	1.5	39	1 012.4	24	104	2 958 8	21.8	
Heavy .	...	2.8	...	...	3.8	...	...	1.3	...	...	0 05	...	...	...	...	...	7.95	...	
Total . . .	9	491.9	11.4	25	614.4	25.3	30	748.8	24.9	2	438.55	2.8	157	1 671.4	58.4	223	3 960.05	34.9	

Number of train-miles: 34 734 825.

Number of English ton-miles: 14 668 095 500.

Number of fractures. { total: 223.
per 10 000 000 tr.-km. or 6 250 000 train-miles: 40.
per 1 billion tkm. or 612 000 000 English ton-miles: 9.

	Percentage of fractures in the part		NUMBER OF FRACTURES:				
			on straight lines or curves of > 800 m. (40 chains) radius	on curves of ⩽ 800 m. (40 chains) radius		on a rising or falling gradient	
	covered by the fishplates	clear of the fishplates		Lower rail.	Higher rail.	⩽ 10 mm. per m. (1 in 100)	> 10 mm. per m. (1 in 100)
D. *Light rails* . .	18 %	82 %	75	24	20	42	33
Medium rails . .	80 %	20 %	94	7	3	55	7
Heavy rails .	...	...	...	...	...	...	...
	Total . . .		169	31	23	97	40
	Miles of single track of each class.		3 469.25	495.80		2 568.7	316.9

		Percentage	
		Light rails.	*Medium rails.*
E. *a*) New clean fractures	with internal transverse fissure	16 %	10.5 %.
	without internal transverse fissure	48.7 %	21 %.
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head	in the foot	12.6 %	11.5 %
	in the head	15.1 %	19.5 %
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head . . .	in the web . . .	7.6 %	37.5 %.

d) Number of pieces rails are broken into:	Number	
	Light rails.	*Medium rails.*
in 2 pieces	110	66
in 3 —	5	21
in 4 —	1	11
in 5 —	...	4
in 6 —	2	2
in 12 —	1	...

(*) Running track only

NAMES OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	AGE OF RAILS : Less than 5 years.			5 to 10 years.			10 to 15 years.			15 to 20 years.			More than 20 years.			The whole of the rails.			*Maximum axle load.*
	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Paris-Lyons-Mediterranean Railways (*).		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
A. Rails outside tunnels. *Light* .	...	68.4	...	...	221.2	...	...	50.3	...	...	184.5	...	204	3 400.8	37.26	204	3 925.2	32.29	19.6
Medium .	8	1 153 9	4.30	30	746.3	24.97	16	309.4	32.12	21	891.2	13.30	212	2 059.6	63.97	287	5 250.1	33.96	
Heavy .	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	
Total . . .	8	1 222.3	4.06	30	967.5	19.26	16	359.7	27.63	21	1 165.7	11.19	416	5 460.1	47.34	491	9 175.3	33.18	
B. Rails in tunnels. *Light* . .	...	2.5	...	...	3.1	...	...	3.1	...	...	9.9	...	10	26.7	232.55	10	45.3	136.98	
Medium	4	42.9	57.97	20	23 0	540.53	15	11.8	789.47	7	27.4	159.09	7	42.3	102.94	53	147.4	223.62	
Heavy .	...	46.0	...	2	18.0	68.96	...	...	...	...	...	...	...	...	...	2	64 0	19.41	
Total . .	4	91.4	27.21	22	44.1	309.86	15	14.9	625.00	7	37.3	133.33	17	69.0	153.153	65	256.7	157.38	
C. The whole of A and B. *Light* .	...	70.9	...	...	224 3	...	...	53 4	...	...	194.4	...	214	3 427.5	38.79	214	3 970 5	33.49	
Medium .	12	1 196 8	6.28	50	769.3	40.38	31	321.2	59.96	28	1 008.6	17.25	219	2 101.6	64.75	340	5 397.5	39.14	
Heavy .	...	46.0	...	2	18.0	68.96	...	...	...	...	...	...	...	...	...	2	64 0	19.41	
Total . . .	12	1 313.7	5.67	52	1 011.6	31.94	31	374.6	51.40	28	1 203.0	14.46	433	5 529.1	48.66	556	9 432.0	36.63	

Number of train-miles : 69 935 370.
Number of English ton-miles : 29 328 205 000.

* Excluding points and crossings.

Number of fractures { total : 556.
per 10 000 000 tr.-km. or 6 250 000 train-miles : 49.40.
per 1 billion tkm. or 612 000 000 English ton-miles : 11.59.

	Percentage of fractures in the part covered by the fishplates	clear of the fishplates	NUMBER OF FRACTURES : on straight lines or curves of > 800 m. (40 chains) radius	on curves of ⩽ 800 m. (40 chains) radius: Lower rail.	Higher rail.	on a rising or falling gradient: ⩽ 10 mm. per m. (1 in 100)	> 10 mm. per m. (1 in 100)
D. *Light rails* . . .	53.74	46.26	111	76	27	172	42
Medium rails .	86.18	13.82	294	37	9	319	21
Heavy rails . . .	100.00	...	2	...	...	2	...
		Total . .	407	113	36	493	63
	Miles of single track of each class.		7 121.0	2 311.0		7 918.3	1 513.7

		Light rails.	*Medium rails.*	*Heavy rails.*	Number of pieces rails are broken into :	*Light rails.*	*Medium rails.*	*Heavy rails.*
E. *a)* New clean fractures	with internal transverse fissure	12.2 %	5.3 %	0 %	2 pieces	170	242	2
	without internal transverse fissure	41.5 %	20.5 %	0 %	3 "	35	74	...
b) Fractures with much rusted old part, extending to the outer surface of the foot or the head	in the foot	26.9 %	31.8 %	50.0 %	4 "	6	17	...
	in the head	12.2 %	29.1 %	100.0 %	5 "	2	4	...
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head . . .	in the web	17.0 %	29.8 %	0 %	6 "	1	3	...
					7 "	...	...	...

OF ADMINISTRATIONS AND DESCRIPTION OF RAILS.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Number of fractures.	Length of single track of this class.	Number of fractures per 1 000 km. or per 625 miles.	Maximu axle loa
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Paris-Orleans Railway. (*)		Miles.			Miles.			Miles.			Miles.			Miles.			Miles.		English tons.
A. Rails outside tunnels. *Light* . .	...	35.1	...	...	341.3	...	...	368.3	...	8	177.6	28.00	16	2 565.8	3.87	24	3 488.1	4.27	18.8
Medium .	1	773.6	0.80	17	1 099.1	9.12	2	178.9	6.95	2	193.1	6.43	32	910.8	21.83	54	3 155.5	10.63	
Heavy. .	...	1.0	...	...	...	...	...	0.15	...	...	...	...	...	...	...	...	1.15	...	
Total . . .	1	809.7	0.76	17	1 440.4	7.33	2	547.35	2 27	10	370.7	16.76	48	3 476.6	8 58	78	6 644.75	7.30	
B. Rails in tunnels. *Light* . .	...	...	...	...	0.1	...	...	0.7	...	...	0.8	...	...	6.2	...	...	7.8	...	
Medium .	...	6.0	...	2	2.6	476.19	...	1.0	...	...	3.7	...	...	5.6	...	2	18.9	65.57	
Heavy. .	1	14.0	44.44	...	42.8	...	9	10.0	562.50	...	...	...	...	...	...	10	66.8	93.02	
Total . . .	1	20.0	31.05	2	45.5	27.28	9	11.7	473.40	...	4.5	...	...	11.8	...	12	93.5	79.74	
C. The whole of A and B *Light* . .	...	35.1	...	...	341.4	...	...	369.0	...	8	178.4	27.87	16	2 572.0	3.86	24	3 495.9	4.26	
Medium .	1	779.6	0.79	19	1 101.7	10.71	2	179.9	6.90.	2	193.9	6.31	32	916.4	21.68	56	3 174.4	10.96	
Heavy. .	1	15.0	41.66	...	42.8	...	9	10.15	552.14	...	...	...	...	...	. .	10	67.95	91.56	
Total . . .	2	829.7	1.49	19	1 485.9	7.94	11	559.05	12.22	10	372.3	16.56	48	3 488.4	8.55	90	6 738.25	8.29	

Number of train-miles: 38. 239 950.
Number of Engl. ton-miles: 19 012 427 000.

Number of fractures { total: 90.
per 10 000 000 tr.-km. or 6 250 000 train-miles: 14 62.
per 1 billion tkm. or 612.000 000 Engl. ton-miles: 2.89.

	Percentage of fractures in the part		NUMBER OF FRACTURES :				
	covered by the fishplates	clear of the fishplates	on straight lines or curves of > 800 m. (40 chains) radius	on curves of ⩽ 800 m. (40 chains) radius		on a rising or falling gradient	
				Lower rail.	Higher rail.	⩽ 10 mm. per m. (1 in 100)	> 10 mm. per m. (1 in 100)
D. *Light rails* . . .	33.33 %	66.67 %	16	8	...	11	8
Medium rails . .	46.43 %	53.67 %	41	5	10	19	11
Heavy rails . .	100.00 %	...	8	2	...	7	3
		Total . . .	65	15	10	37	22
	Miles of single track of each class.		5 406.0	2 332.25		4 729.2	1 107.0

		Percentage		
		Light.	*Medium*	*Heavy.*
E. *a*) **New clean fractures**	with internal transverse fissure	4.17	1.79	...
	without internal transverse fissure	41.66	19.64	...
b) **Fractures with much rusted old part, extending to the** outer surface of the foot or the head	in the foot	20.83	8.93	...
	in the head	25.00	21.43	...
c) Fractures with much rusted old part, *not* extending to the outer surface of the foot or the head . . .	in the web	8.34	48.21	100

d) Number of pieces rails are broken into	*Light rails.*	*Medium rails .*	*Heavy rails .*
2 pieces	20	50	9
3 —	2	6	...
4 —	2	...	1
5 —	...	...	...
6 —	...	...	...
7 —	...	...	...
8 —	...	...	...
9 — and over. . . .	...	...	...

* Running track only.

(To be continued.)

NEW BOOKS AND PUBLICATIONS.

[385 (.460)]

El Problema de los Ferrocarriles Españoles, 1933. (**The Spanish Railway Question**, 1933.) — One volume (10 × 8 inches), of 194 pages, with many statistical tables and diagrams. Madrid, May, 1933; Grafica Administrativa, Rodriguez San Pedro, 32. (Price : 10 pesetas.)

This work is a fully documented study of the crisis through which the Spanish railways are passing. The problem is first of all sketched, in an introduction, under three aspects : juridical, financial and economic. The volume itself is divided into two parts : the first is devoted to a methodical examination of the various elements characterising the Spanish railways; it gives statistical details relating to these railways, and at the same time compares these figures with those relating to foreign railways. The first chapter shows : the progressive growth of the system, with the main features of the gradients; the development of electrification; the position as regards motive power and rolling stock. The second chapter deals with questions relating to the staff : total numbers and numbers per unit of trafic; position of the staff as regards remuneration and insurances. Data relating to operation : passenger and goods traffic, rates, receipts, analysis of expenditure, operating coefficients, fiscal charges, capital invested in the railways and its earnings, are dealt with in the succeeding chapters.

The second part deals with the railway problem as seen at the present time in Spain, and considers possible solutions. The theory developed can be stated as follows : the railways of Spain at the present time fully meet the requirements of the national economy except as regards return on the capital invested. The work tends to show that the development of the Spanish railways is quite sufficient to meet the traffic requirements, especially when account is taken of the growth of road-transport; the problem therefore only lies in the direction of improving the existing lines : conversion to double track, renewal work, etc.; the motive power and rolling stock also meet the requirements; the staff has reached a high standard of living; the railway industry no longer uses stores other than those produced in the country; the railway forms a source of considerable revenue for the public treasury; the Spanish rates are relatively lower than those of most other countries; in spite of the increase of expenditure and the very high taxation, the operating coefficient remains satisfactory. The position of the railways is therefore healthy; but in opposition to these favourable features, the problem of the insufficient return on the capital invested, the essential cause of which lies in the too low level at which the rates have been maintained, still remains. This insufficient return prevents the investment of new capital, which is indispensable to maintain the railways in the highest state of efficiency. As causes, must be mentioned especially those affecting the financial equilibrium of the operating services; the economic depression, road motor competition, the increase of expenditure as a result of social and economical legislation, the taxation on transport, about the highest in Europe, and finally the increasing intervention of the State in the regulations. The above are the subjects dealt with in the first chapter.

The second chapter considers the possible solutions : State management; private management, and joint management, as in force at the present time on the Spanish railways. The advantages and drawbacks of the two first solutions are discussed, the discussion being based upon the experience of recent years in other countries. The joint regime, with the State and the companies who have the concession, has been in force in Spain since the passing of the law of 1924; it involves the management by the companies, with the rates so fixed that the railways can meet all operating costs, social insurance, financial and fiscal charges, while securing a normal return on all invested capital, the management being controlled by the State which, in principle, assures the investment of new capital. These legislative arrangements have apparently not been carried out properly, especially as regards revision of rates, and this has resulted in the present difficulties.

The work terminates by suggesting that, whatever form of management be adopted, a series of measures be adopted for improving the position, and in particular : the revision of the social legislation, much too strictly enforced at the present time; the reduction of the transport tax; the reform of the administrative regulations; the suspension of the construction of new lines decided in recent years; and finally the regulation of road transport and its co-ordination with the railways.

[385.]

SAITZEW (Manuel), Professor of Political Economy in the Law School of the University of Zurich. — **Mission et régime des chemins de fer dans l'économie nationale.** Contribution à l'étude du problème de la concurrence entre le chemin de fer et l'automobile. **(The mission and régime of railways in the national economy. — A contribution to the study of the problem of competition between rail and road.)** — One pamphlet (9 1/2 × 6 1/2 inches) of 88 pages. — 1932, Berne, Librairie Edition, S. A.

Switzerland is one of the countries in which road motor competition has been most heavily felt. At an early period, methods for diminishing the disastrous effects thereof were investigated and steps taken to co-ordinate transport by rail and by road. Recently again new legislation has been proposed by the General Management of the Federal Railways.

A study by a Swiss author, written with the object of bringing out the specific characteristics of railways and their economic role, their importance from the social aspect, their own rates structure, and their public and national utility, could not fail to be of great interest. Although the author may have examined more especially the position of the Swiss railways, noting and underlining the facts peculiar to them, it may be said that the whole work applies equally well to the railways of other countries.

While declining any intention of making the question of road motor competition the principal matter, which would involve other developments, he has taken care, basing his arguments on well established facts and upon incontestable statistical data, to prove that the position taken by the railways in the economic field is such that there is an absolute necessity for the State to take such action as will assure their survival.

We will mention briefly, among the ideas developed in this work, the great influence exercised by the railway on the material life of the nation by the multiplication of transports, the importance of the capital invested in building them, the frequent intervention of the State in railway matters, justified by the great interests involved, and which often has placed very great burdens on the railways, certain of which are outside their proper scope. The chapter dealing with the relationship to be established between the cost of transport and the price the customer is asked to pay merits special attention. The principle of fixing the rates according to the specific value of the goods, a valuable principle which enables the railways to transport materials of little value but essential to industry and all economic life, is very clearly defined and defended : the whole rates structure is the result of this principle. The competition of the road motor by easily cutting into the higher rated classes of goods thereby runs a risk of ruining the necessary equilibrium of the commercial organisation of the railway.

The author, in consequence, comes to the conclusion that measures of the broadest character are now indispensable and that none but the political authorities of the country can take them. He indicates two solutions and discusses their consequences. The first consists of operating the railways according to the principles ruling private industry, at the same time exempting them from the burdens and obligations which have been imposed upon them for the public good. He rejects this solution on account of the unfavourable results it would necessarily have upon the national economy as a whole. The second would be to restore the situation such as it was before the road motor appeared, by incorporating the latter into a transport organisation managed by the State in the general interest. By this method an endeavour would be made to establish reasonable collaboration between the railway and its competitor. Without entering into the details of the methods of application, the author limits himself to indicating the possibilities of regulating transport by lorry, and points out that one of the first measures would be to deprive the road motor of the privileges it possesses today.

E. M.

[385. (09. 2]

OBITUARY.

The Right Hon. Viscount CHURCHILL, G. C. V. O.,

Chairman of the Great Western Railway Company, (Gt. Bn.),
Life Member of the Permanent Commission of the International
Railway Congress Association,
President of the Xth Congress (London, 1925) of this Association.

We have heard with the deepest regret of the sudden death, on January 3rd., or Viscount Churchill, Chairman of the Great Western Railway Company.

Born in 1864, Lord Churchill was educated at Eton and at the Royal Military College, Sandhurst; he served with the Coldstream Guards from 1884 to 1889. He succeeded his father as Baron Churchill in 1886 and, in 1902, was made a Viscount. He was Lord Chamberlain at the Coronation of King Edward VII and Master of the Robes at the Coronation of King George V.

In 1905, he joined the Board of Directors of the Great Western Railway Company and succeeded the late Mr. Alfred Baldwin, father of Mr. Stanley Baldwin, the former Prime Minister, as Chairman of the Company in 1908. Under his wise guidance, incessant improvement rather than extension was the policy of the Great Western Railway. He fully realised that it was the duty of a Board to direct and not to interfere in details, and his most cordial and encouraging attitude towards the staff resulted in developing their initiative and carrying the undertaking forward to a yet greater efficiency.

Lord Churchill was Chairman of the British Railway Companies' Association from 1920 to 1922. His term of office included the critical period culminating in the passage of the Railways Act, 1921. The reshaping of this Act, so as to make it generally acceptable to the Companies, their staff and the public was largely due to his skill, courtesy and hard work. As a tribute to his services in this respect, the Railway Companies' Association presented him, in 1925, with his portrait in oils, by Sir William Orpen. In 1931 and 1932, he was again Chairman of the Railway Companies' Association.

During his Chairmanship of the Great Western Railway Company, which lasted over 25 years, he had experience of no less than five General Managers, all Great Western trained.

He was most assiduous in the attention to his duties; he was always most approachable and the staff were all devoted to him.

Viscount Churchill was the holder of a number of honours and decorations. He was also Chairman of the Fishguard and Rosslare Railways and Harbours Company and of the British Overseas Bank Limited, a Director of the Grand Union Canal Company, of the Peninsular & Oriental Steam Navigation Company and of the British India Steam Navigation Company.

Lord Churchill had been a member of the Permanent Commission of the International Railway Congress Association since 1910, and was the President of its Xth Session (London, 1925), on which occasion he was made a life member of the Permanent Commission. He always took a keen interest in the work of the Association. His death will be keenly felt by his former colleagues of the Railway Congress Association and, on their behalf, we wish to convey our sincere sympathy to his relatives and to the Great Western Railway Company.

The Executive Committee.